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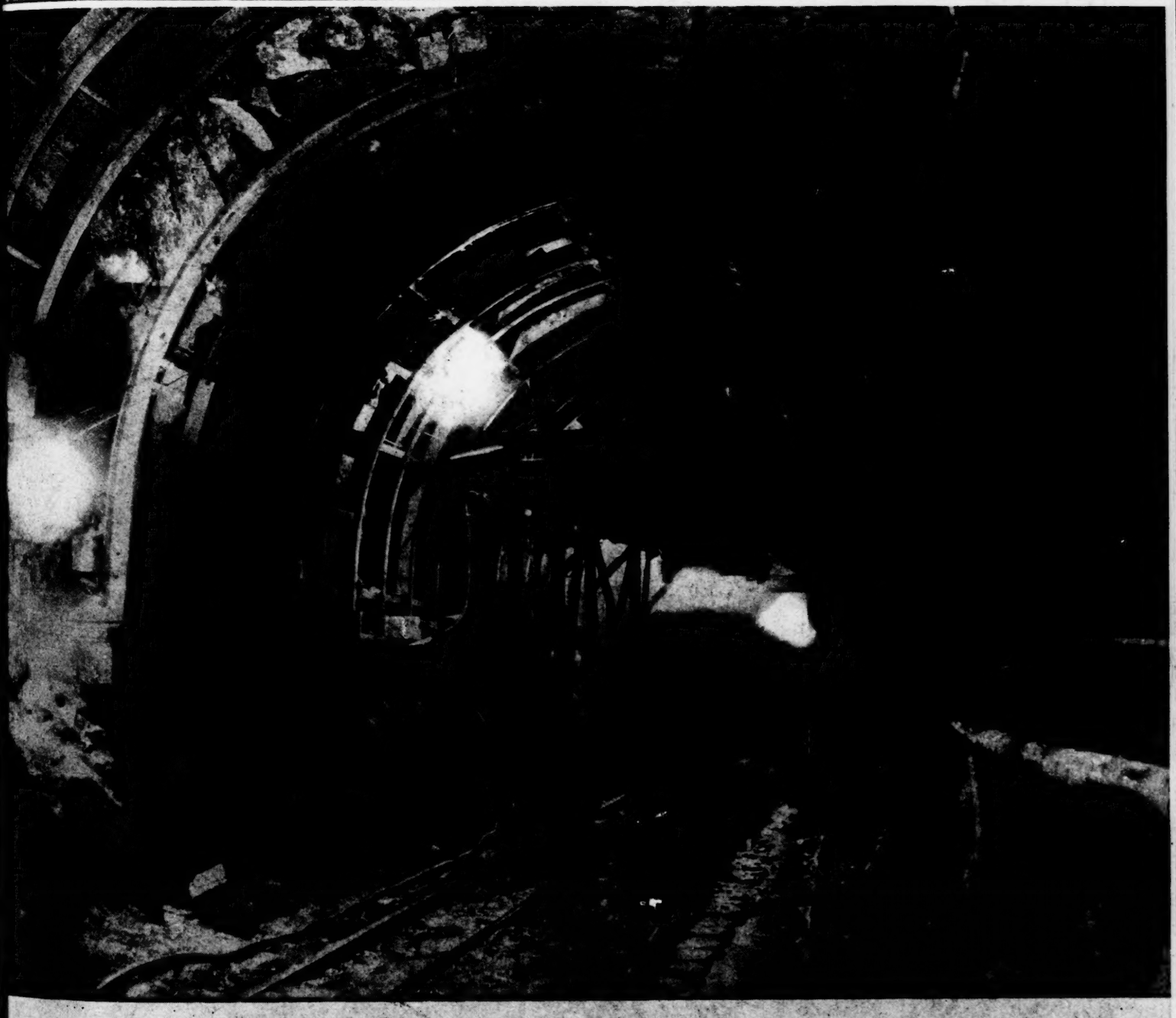
HEATING AND VENTILATING

AIR CONDITIONING, REFRIGERATION, PIPING, AIR SANITATION, HEATING AND VENTILATION



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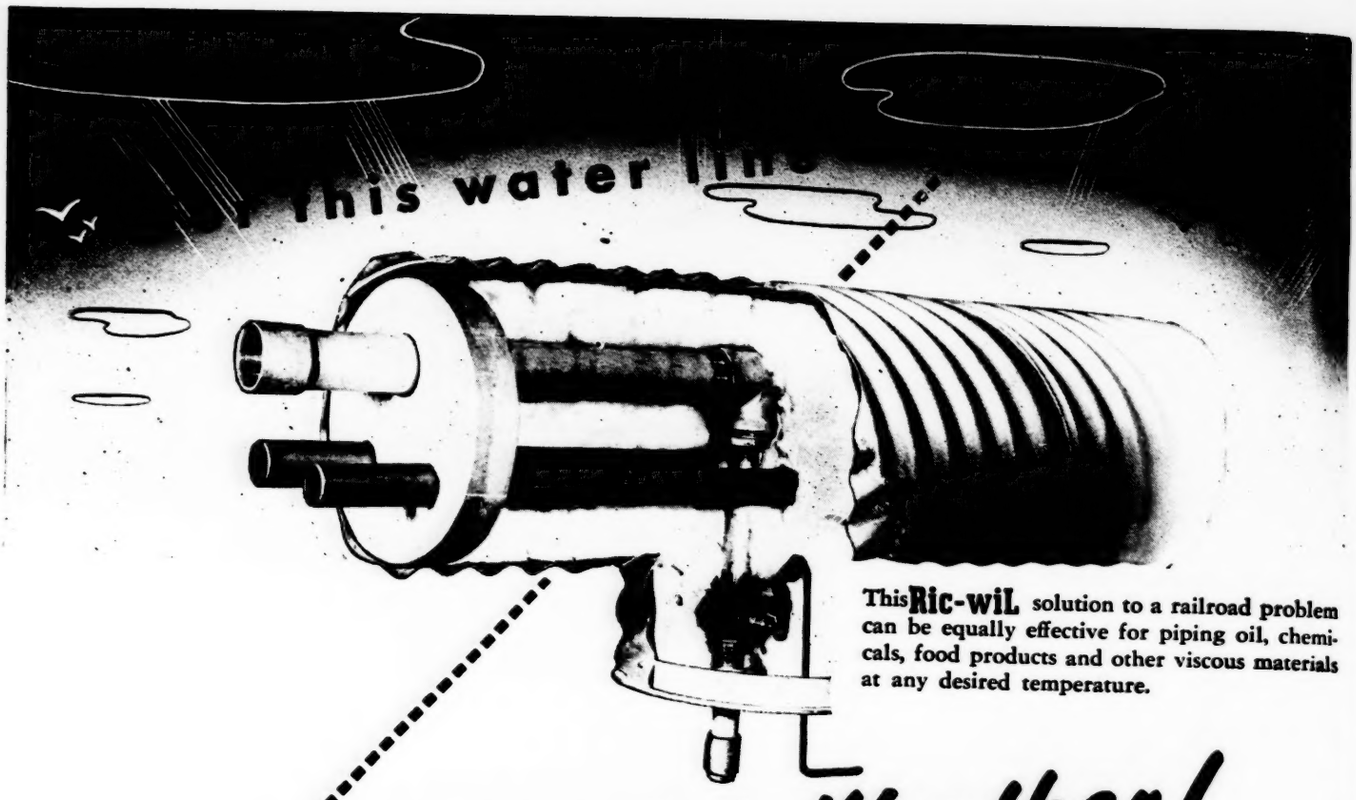

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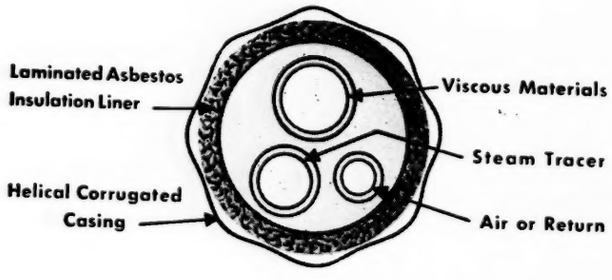

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A MONTHLY MAGAZINE FOR ENGINEERS AND CONTRACTORS CONCERNED WITH THE DESIGN AND APPLICATION OF AIR CONDITIONING, REFRIGERATION, PIPING, AIR SANITATION, HEATING AND VENTILATION IN BUILDINGS

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PUBLISHERS OF MACHINERY & MACHINERY'S HANDBOOK
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THIS MONTH'S COVER

Ventilation of the Delaware Aqueduct during construction had to fight heavy moisture, methane, high pressure, and rock dust. W. B. Harris tells how the problems were solved beginning page 63.

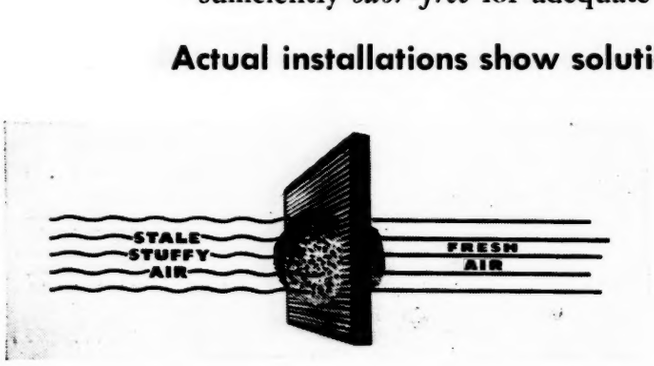
Three air conditioning problems that Air Recovery can solve

1. When the heating and cooling equipment cannot handle enough ventilating air to supply the fresh air demand.

2. When adequate ventilation requires bringing in large volumes of outdoor air *which has to be conditioned at considerable cost.*

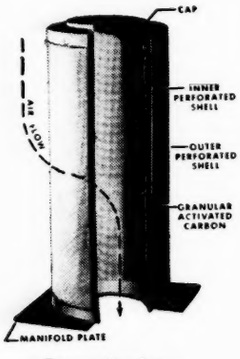
3. When it's difficult to bring in outdoor air or when the intake air isn't sufficiently *odor-free* for adequate ventilation.

Actual installations show solutions—demonstrate how Air Recovery cuts costs

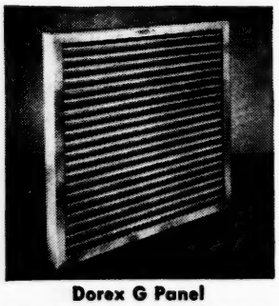


It works like a sponge. The air circulates through beds of specially processed Dorex activated carbon, which literally soak up the most minute traces of any odors originating from occupants and their activities, foods, beverages, tobacco smoke, machines and kindred sources. Thus, recirculated air, already heated or cooled, is restored to original freshness, making it acceptable for human consumption or industrial processes. Average resistance to air flow added by Dorex Units is about 0.15″ water gauge, a negligible percentage of a system.

Canister arrangement



Dorex Canister



Dorex G Panel

Dorex types and sizes have been developed to fit all standard air conditioning installations. G panels are used where space is limited and in unit air conditioners. Canisters are installed in the duct of central station type of air conditioning systems.

IN THIS MAGAZINE in Mar. '47, we described how the majority of railroad companies solved the first condition mentioned above. In May, we reported how the second condition was overcome in the Criminal Courts Building, New York. In the present case, the engineers improving the air conditioning for the First National Bank Building, Detroit, were faced not only with the third condition, an intake air odor problem, but with the first as well—the cooling equipment could not handle enough ventilating air to supply fresh air demand.

The solutions to these problems illustrate where, how, and why you can profitably design Dorex Air Recovery Equipment into similar high occupancy jobs.

Even though the outdoor air intake of the First National Bank Building is located at what is ordinarily a good position on the fourth floor level, it was befouled under certain wind conditions by kitchen exhausts from several neighboring restaurants. In addition, business in the bank had increased. Added personnel and customers created a ventilation demand that overtaxed the existing conditioning equipment to a point where the operators had to choose between desirable temperature and proper ventilation. At times, it was necessary to bring in a maximum amount of air for dilution purposes. You can imagine the spot they were in when cabbage smells were coming their way.

The application of Dorex in a case like this is simple. Since it is a problem of ventilation—and that's a problem of odors—remove the odors. That's just what the engineers did.

They installed enough activated carbon canisters to purify 20,000 CFM of return air and 10,000 CFM of outdoor air. The 30,000 CFM of fresh air provided ample ventilation. (See before and after installation sketches.)

Since the existing conditioning equipment could handle the 10,000 CFM of outdoor air easily, this solution not only eliminated the need for additional conditioning equipment, but saved on operating expenses as well. In February, before the first heating season with the Dorex equipment was over, Merle Bennett, the Chief Engineer of the First National Bank, reported that he had already saved 108 tons of coal and

Washington News

LORING F. OVERMAN

WASHINGTON'S major happening during June—passage of the Taft-Hartley labor law over the President's veto—in its articulation of aroused public opinion embodies a warning to the construction industry. In the case of the building industry, including its heating, ventilating and air-conditioning segment, the warning does not carry with it any threat of new legislation. Instead, it promises only to invoke the oldest of laws—that of supply and demand.

In a word, the public—and it is not calling upon Congress to implement its decision—is saying: "Mr. Construction Industry, your prices are too high. If something doesn't happen to bring them down, we'll just put our dollars back in our pockets, and roll up the blue prints and wait."

Demand – Yes, but:

Only half of the picture has been seen when it is concluded that a building boom is inevitable because there was one after the last war, and because the *need* following the recent one is infinitely greater. Building costs were comparatively low in comparison with rental rates during the 20's—an ideal business stimulator from the standpoints of prospective individual home owners and builders of rental properties.

This relationship has been reversed in the current situation, with building costs skyrocketing since 1945, and rents held more or less in line by Government controls. With these controls scheduled to continue until March, 1948—subject in certain cases to "voluntary agreements" to include 12-month leases and permitting increases of not more than 15%—there follows a condition which market analysts have classed as "a perfect setting for a stalemate in construction." This, they say, brings out the other side of the picture, with millions of prospective builders of both housing and commercial properties deciding that it costs too much to avoid "doubling up" in homes, or to unroll that business blueprint at this time.

Housing Warnings

Since stress has been placed on housing since V-J day, the importance of this field as an indicator cannot be overlooked by the heating, ventilating and air-conditioning industries, even by those whose major interests lie with commercial and industrial construction. The Department of Commerce — which earlier estimated a housing construction volume of $6 billion for this year—currently predicts a volume of only between $4.25 billion and $4.8 billion.

Permits for all urban construction in the first three months of this year fell to $923 million, a drop of $515 million from the total for the same period last year. Construction contracts awarded in the 37 States east of the Rocky Mountains dropped 18% in April of this year below the volume for the corresponding months of last year.

All this, despite the fact that demand is overwhelming; supply inadequate! What can the trouble be? Perhaps labor is short?

Employment Dropping

Although it had been forecast that labor would prove the real bottleneck of the construction situation, the Bureau of Labor Statistics reports that on-site employment in new private residential construction in April of this year was 383,000 in comparison to 406,000 in the same month last year—a drop of 23,000. It would appear that the construction slowdown is not entirely due to a lack of labor—although it is highly probable that labor, too, is pricing itself out of the market.

About Non-Housing

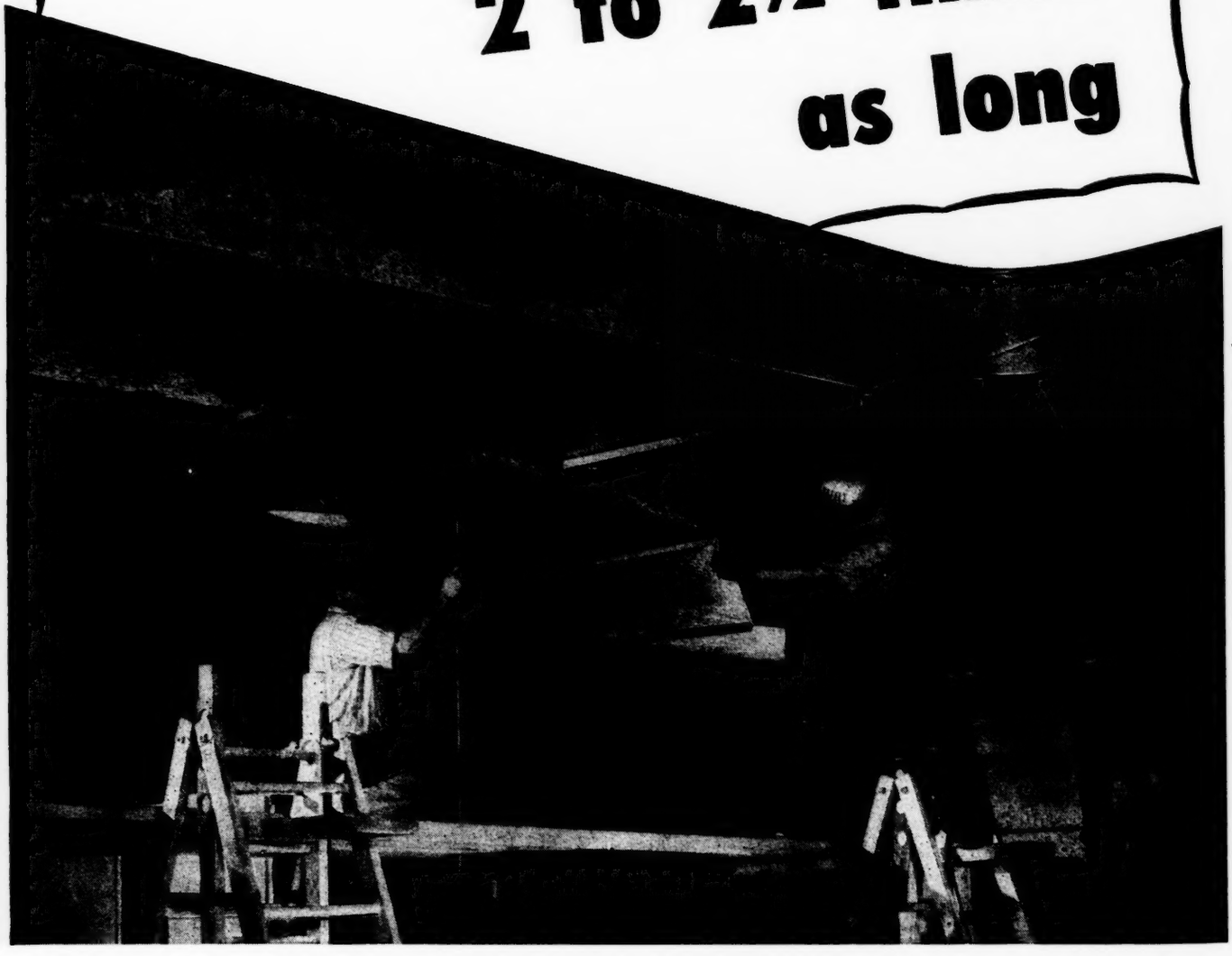
There are those who have contended that the housing program, with its emphasis on veterans' homes, has interfered with, rather than aided, the construction total. Congress, as this column went to press, appeared to accept this view, and it appeared that President Truman would have to sign a House and Senate compromise bill relaxing housing controls in order to salvage rent control.

As agreed to in conference, the bill repeals all of the Patman emergency housing act except: The Office of Housing Expediter is continued until February 29, 1948; previously granted allocations, priorities, and the price limitations connected therewith continue in effect; construction permits can no longer be required except for amusement and recreation facilities; market guarantees for factory-built homes are continued; homes built for sale or for rent must be reserved for veteran purchasers for 30 days after completion; and Section 603(a) of the National Housing Act, providing for liberalized construction loan guarantees for rental housing, is continued until March 31, 1948.

Tied to this bill, which gives the construction industry permission to go ahead virtually without restraints, is tied a modified rent control bill also intended to encourage new construction. Rent control is continued until February 29 but is modified in some respects. New construction completed after February 1, 1947 is specifically exempted from rent control, except for the retention of rent ceilings on priority-aided construction. Also exempted are hotels, motor courts, and other transient accommodations.

Whenever, after January 1, 1948, landlord and tenant voluntarily negotiate a lease extending to at least December 31, 1948, an increase in existing rates of not more than 15% will be valid. The Housing Expediter is authorized and directed to remove rent controls before February 29, 1948 in any area where, in his judgment, the need for them no longer exists.

Beth-Cu-Loy Sheets last 2 to 2½ times as long



Ordinary air ducts made from Beth-Cu-Loy Galvanized Sheets can be expected to last twice as long as those made from plain galvanized steel sheets—or even longer.

The principal reason for this added service is the resistance to rust provided by Beth-Cu-Loy's copper-bearing steel base. It is manufactured from open-hearth steel that contains 0.20 to 0.30 pct copper. In atmospheric-exposure tests by the American Society for Testing Materials this composition has proved itself more than twice as corrosion-resistant as ordinary steel. Further rust-protection is given to Beth-Cu-Loy Sheets by the brightly spangled, uniform coating of Prime Western zinc.

Aside from their lasting qualities, Beth-Cu-Loy Sheets are easy to form, to seam and to solder. They make fast work of difficult jobs. Their slight additional cost is more than balanced by their longer life and other advantages.

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Beth-Cu-Loy Galvanized Sheets

BETHLEHEM STEEL

With the green light given practically all types of projects, perhaps the answer to the slowdown will appear! But the recently-released Civilian Production Administration survey, through February, shows that more than $165 million in non-residential construction authorized between March and September of 1946, has been deferred by the sponsors. Of that amount, 62% was put off or cancelled because of "excessive costs" or because "contractors were unwilling to set prices."

In a limited sampling of 89 projects in the Washington, D. C. area, authorized between March and September of last year, 21 projects to cost $5,245,000 had not been started by March of this year. Of this number, sponsors of 10 projects to cost $3,323,000 had either definitely abandoned the projects or had postponed them until after July 1 of this year.

$5,000 Per Unit?

Despite the fact that the majority of prospective home owners desire homes priced under $5,000 (or rental units based on that figure), the Senate Banking and Currency Committee took action during June that virtually admits officially that a $5,000 unit is a thing of the past. To speed plans for low-rent public housing projects, it recommended that Congress relax a 10-year-old regulation setting $5,000 as the maximum that could be spent on a dwelling unit built with Federal funds. The committee approved a bill that would permit local housing authorities to pay the difference between $5,000 and present production costs.

On the same day, June 17, the House disapproved President Truman's reorganization plan to combine all of the Government's housing bureaus into a single Federal housing and home finance agency. The plan is now before the Senate for consideration, and unless specifically rejected within 60 days, becomes effective. Nearness of adjournment is causing some to wonder whether action can be taken before the deadline.

The United States Chamber of Commerce observes that "the plan is diametrically opposed to Chamber policy in that it would combine the administration of the Federal home financing agencies with both war liquidation activities and subsidies for Government housing activities. The danger in this is the strong probability that sound and reasonable principles of credit will be weakened by attempts to adapt them to the ends of social welfare, resulting in the deterioration of a tested system of home financing."

Congressional Probe

Meanwhile the House Labor Committee, on June 22, ordered "an investigation of racketeering and monopoly in the housing and construction industry." Representative Ralph W. Gwinn, New York Republican, who was designated by Chairman Hartley to make the investigation, told newsmen it would delve into "what appears to be a move to bring about collectivism through increased Federal Housing programs. Chairman Hartley said the sub-committee would investigate "material and labor costs and questionable practices relating to the economics of housing and construction generally" and would inquire into "numerous complaints of abuses that are paralyzing the building industry."

"The building trades unions," Gwinn declared, "are tying up all housing, including housing in whole cities and in some instances in States, unless private builders agree to terms calling for wages and overtime that they cannot pay if they want to sell their houses." The practice, he said, has been aimed primarily at moderately priced houses and apartments.

"The unions," he said, "are beginning to clamor for more Federal housing because the Government is the only one able to pay the exorbitant prices, since it can pass the costs along to the taxpayers. Unwittingly, these unions are joining forces with the collectivists in a move to get more public housing, the greatest avenue there is to a collectivist society. Government housing means political part control of those who occupy the housing, and in other countries it always has led to allocation of space under a one-party system."

Question Mark?

Indicative of the varied thinking on the problem is the Washington Post's editorial comment on the pending hearings, to open in September in New York, Chicago, and West Coast cities.

"Whatever may be said of the short-sighted policies of builders and organized building labor, the overshadowing fact is that Congress has not yet given us a national housing policy. It has not yet recognized good housing as one of the first essentials of our national health and welfare. Instead, Congress seems to be moving in the opposite direction."

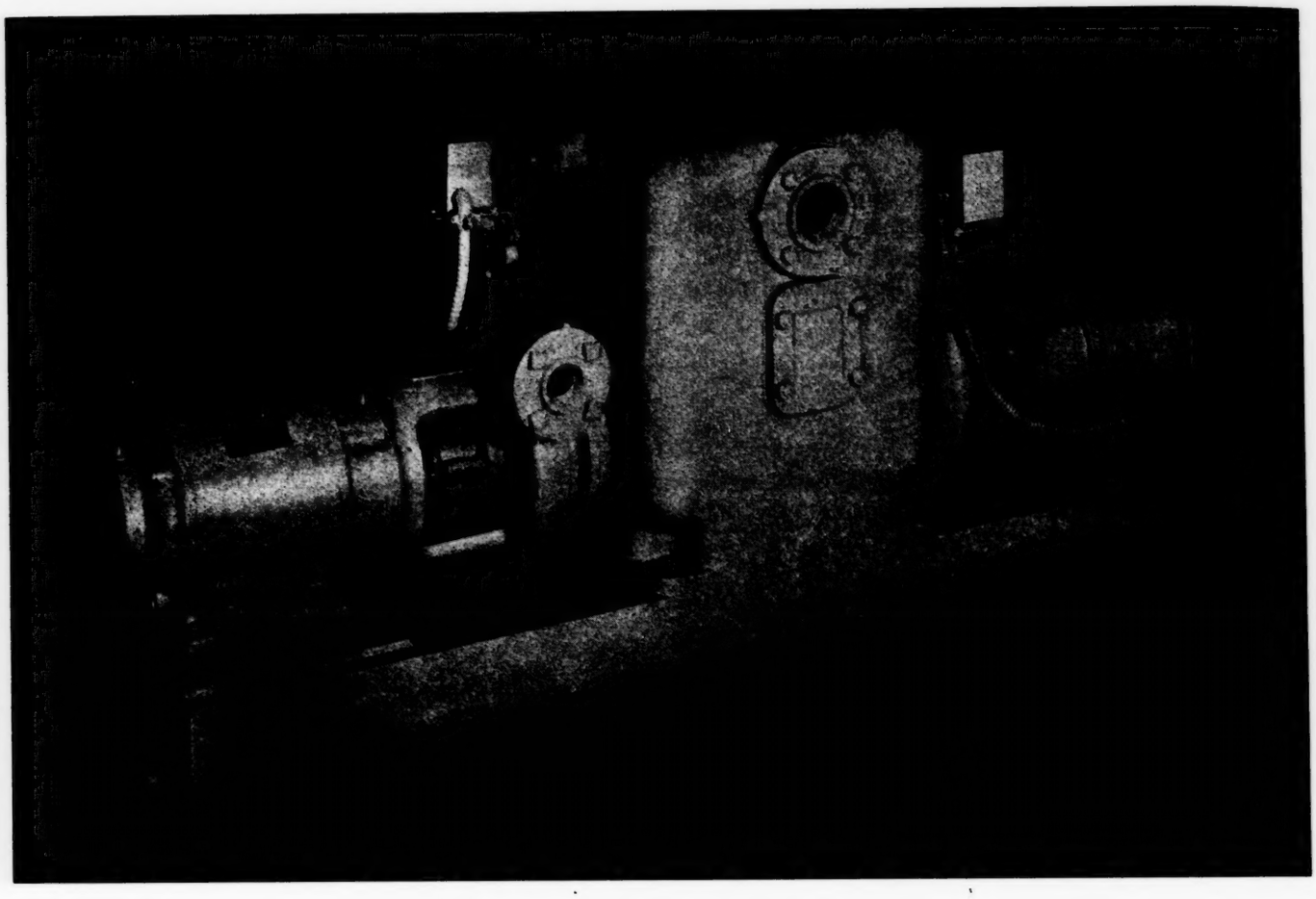
Guidance Needed

And so the arguments go. Both management and labor in the construction field seem most effective in opposition. Elements have made themselves heard against rent control, against pre-fabrication, against slum clearance, against preference for veterans, against the Taft-Ellender-Wagner bill. Opponents of this latter bill label it a socialistic measure—yet one of its sponsors is the same Senator Taft who engineered the business-welcomed Taft-Hartley law over a veto.

Meanwhile — torn between conflicting views — construction is slowing down, demand remains unsupplied, and the market for heating, ventilating, and air-conditioning equipment and services is being compressed between the extremes. Present, most assuredly, is a problem of major significance to every individual whose future is tied in with the construction industry. Problems of such magnitude do not solve themselves.

Needed now is the informed interest of every one in the construction industry and its satelites. Needed now, by Senators and Representatives in Washington, is the best judgment of the industry's best brains. Not pressure — judgment. Retreating labor forces spent millions of dollars for pressure that proved ineffective against an overwhelming public opinion demanding a curb on labor leaders' powers. Today, public opinion is wondering about construction.

MAXIMUM EFFICIENCY FOR STEAM-HEATING SYSTEMS ON HEATING, DRYING AND PROCESS WORK.



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Designed for steam systems on heating, drying, or process work, these pumps offer the most reliable and efficient means of returning hot condensate to boiler or hot well. In many cases these pumps will save costly installation of boilers in a pit to provide gravity flow of returns from basement radiators and heating devices.

The Jennings Condensation Pumps are sturdy and compact in construction, and combine receiving tank, pump and driving motor in a single assembly. Pumps are bronze fitted throughout, with tobin bronze shaft. The pumping impeller is of special design, made possible by our wide experience with return line heating pumps, and is especially adapted for handling hot water with the greatest possible efficiency.

Full automatic control is furnished by means of a ball float and float switch mechanism mounted on the receiving tank. All wiring is made up at the factory, and it is necessary only to connect the pump to the system and hook up the leads. Complete information regarding these economical pumps sent promptly on request.

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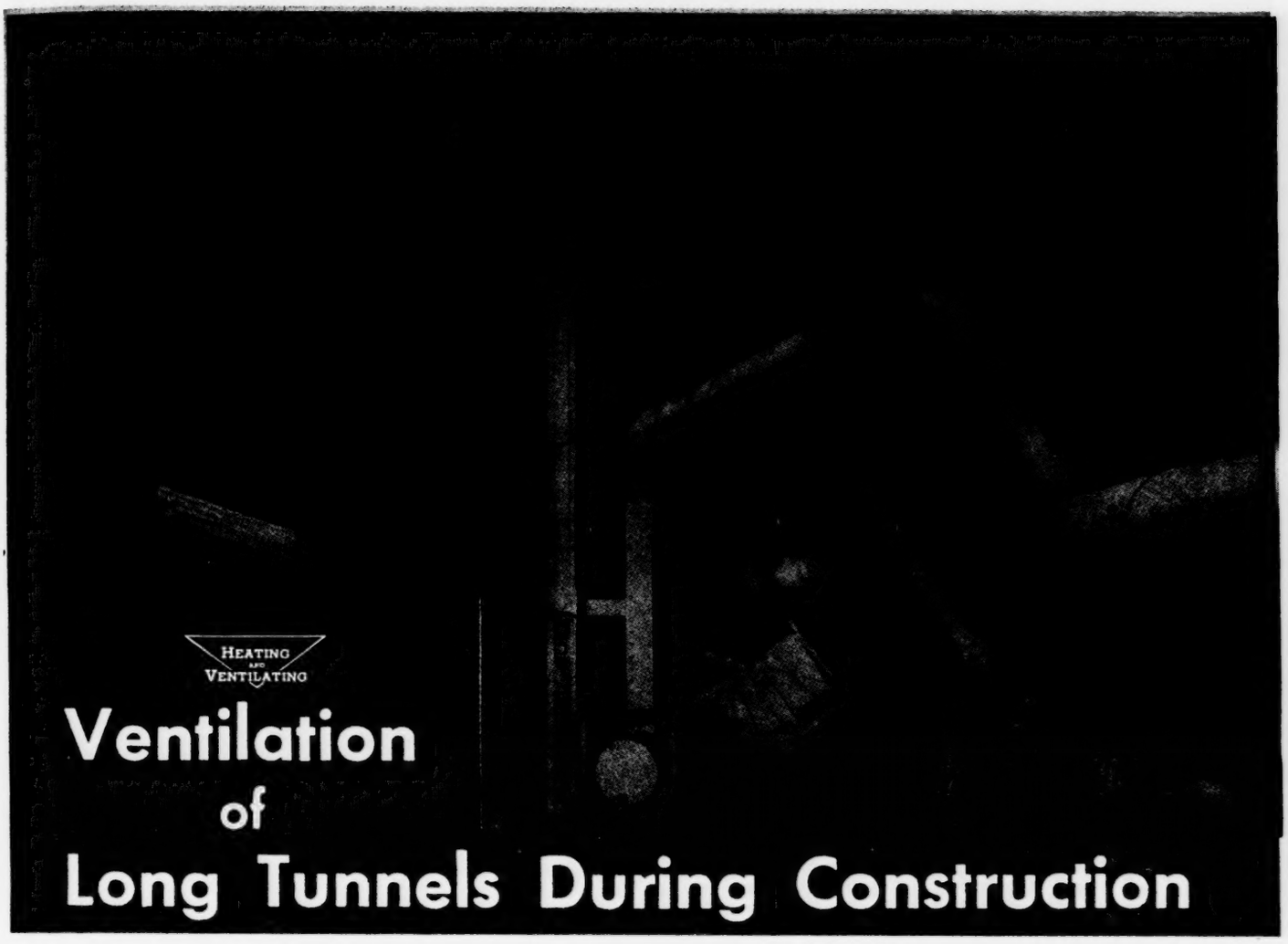

HEATING AND VENTILATING

Ventilation of Long Tunnels During Construction

W. B. HARRIS
Engineer, New York State Department of Labor, Division of Industrial Hygiene

Basic engineering factors which effect the adequate supply of air through a single tube underground operation, are presented in this three-part article. Sound ground work is provided from which engineers can calculate the ventilation of similar tunnels.

MUCH material has been published regarding the necessity and value of ventilation during mining and tunneling operations. From this accumulated mass of data, the reasons and the empirical quantities necessary for maintaining healthful underground conditions are fairly well known. Tebbens and Tabershaw have recently proposed what purports to be an absolute yardstick for the determination of the exact quantity of air per minute necessary to maintain a required dust count.

Data in this article and the conclusions drawn are based upon a study of the construction of the Delaware Aqueduct Tunnel for the New York City water system, from the shaft sinking to the completed cylindrical tunnel, over a period of about two years. During that time Industrial Hygiene liaison was maintained by the author between the interested governmental agencies and all the twelve contractors engaged in driving the various tunnel sections.

The tunnel is about 90 miles long, from 13.5 to 19.5 ft diameter, and an average below-ground level of 400 ft.

It might be well at this point to review briefly the operations involved in driving a long tunnel. A vertical shaft of suitable diameter is sunk to the level at which the tunnel will be driven. At this point a large steel drill carriage is set up, on the front of which is mounted a number of pneumatic drills. In the case of a tunnel of the dimensions noted, 6 to 10 drills were used. A suitable pattern of drilled holes, approximately 10 ft in depth, are then drilled in the direction in which the tunnel is to progress. These holes are loaded with dynamite and properly tamped. The equipment necessary for the previous operations is then withdrawn from the area, perhaps as far back as 1,000 ft. The round is blasted, and after a delay period, of from 15 minutes to one hour, during which the gases from the explosion are removed by means of ventilation, the men return to the heading to remove the blasted rock. This next operation (removing the rock), called "mucking," is carried on by means of a mechanical shovel, feeding the muck back to a belt conveyor which dumps the material into railroad cars. After the material has all been removed from the tunnel, the mucking machine and all the necessary and appurtenant equipment are

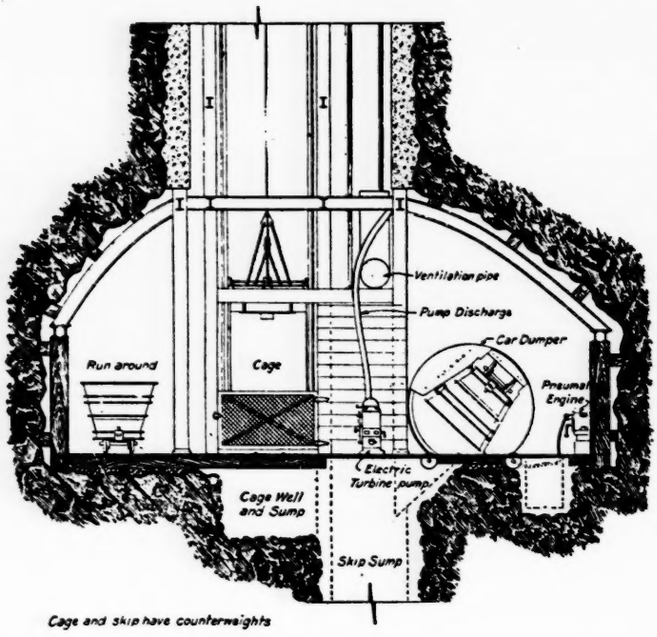

Fig. 1. Section through one of the many shafts required in the construction of long tunnels.

withdrawn and the drill carriage is again placed at the heading for the beginning of the next cycle.

Simultaneously a similar operation is carried on at a convenient distance from the first shaft and a tunnel is driven from the second location toward the operation previously discussed. The two tunnels finally meet at a point approximately midway and the operation of clean-up is begun. This operation consists of removing all equipment, railroad tracks, and debris which were left in the tunnel. In most cases a concrete lining is then poured into the tunnel, making a completed tube of the required shape. Dust is created in the operating area during the drilling operations and during the mucking operations. Gases are created by the blast and persist through the beginning of the mucking operation.

Because of inherent differences, this discussion will be divided into two parts: Ventilation of tunnels which are open at both ends or have been holed through; those which have one dead end or are still being driven toward a meeting. In all but one case, the operations which were studied were tunnels opening to the atmosphere through vertical shafts. The single exception was a diversion tunnel necessary for the construction of a dam. This tunnel was driven through a hill from open portal to open portal. All operations studied were carried on at atmospheric pressure.

Dead-End Operations

Of the two types of operations, by far the more difficult to control, and more important from the standpoint of hazards, is that carried on in dead-end tunnels. Of these, all the major dust and fume producing operations such as drilling, blasting, and mucking are carried on, and all ventilating air must be supplied by means of an extensive piping system.

It is impossible to specify the exact minimum quantity of air which must be supplied to maintain healthful conditions in every underground location and for every set of operating conditions. The variables are numerous and each has a wide range of variation. Methods of drill mounting, speed of drilling, permeability of rock, amount of moisture, are all factors which affect dust production. Type of powder, tamping methods, pattern of drilling, etc., all tend to modify the quantity and type of gas encountered after blasting. However, it was our experience in the operations studied, that the use of 1,200 cfm air per drill, supplied continuously and properly (the drills were of the New York State approved type as specified by Industrial Code), was sufficient to control the dust produced during drilling, and allow a rapid return after blasting (less than 30 minutes). If sufficient water were used on the muck pile, an adequately low dust concentration was maintained during mucking operations.

This air quantity was more effective as it was applied closer to the operating face, but at a distance of from 60 to 90 ft, it was found to be adequate. Carrying the main ventilation pipe to a point closer than 60 ft from the operating face usually led to frequent smashing of end pieces by large chunks of rock being thrown back from the face during blasting. If the terminal end of the vent pipe were at a distance greater than 90 ft, the effectiveness of the control diminished rapidly. Thus, in a usual heading operating 7 to 9 drills drilling about a 16-ft diameter tube, adequate ventilation was provided by supplying about 10,000 cfm of air from a distance of 75 ft.

All installations seen were arranged so that air could either be blown into or exhausted from the operating face. This was accomplished by a suitable blower installed at the surface. Ductwork from this blower was carried down the shaft and along either the tunnel wall or roof directly to the operating area. In one group of installations it was necessary to exhaust continuously because of methane-producing strata. In others, the most usual and normally the most effective method for controlling the hazardous materials, was to exhaust for a period from 10 minutes before until 30 minutes after the blast, and to blow air throughout the remainder of the cycle. Other methods of providing adequate ventilation involved the use of auxiliary air-moving equipment.

Blowing high pressure air directly from the compressed airline used to supply the drills was the least effective and least desirable method of introducing auxiliary air. Because the necessary air quantities are of the order of many thousands of cubic feet per minute, air from this source is usually too small a quantity to be of much value. This method of air supply has the added disadvantage of materially increasing the fog density because of rapid expansion of the compressed air. In every case where compressed air was released into the tunnel atmosphere, a fog cloud formed.

The use of fan equipment mounted directly on the drill carriage was extensive, and of variable benefit. Probably the most effective equipment consisted of a propeller type fan mounted directly on the drill carriage so as to draw the dust-laden air from the face area and discharge it away from the face. This was found to reduce the fog concentration under all con-

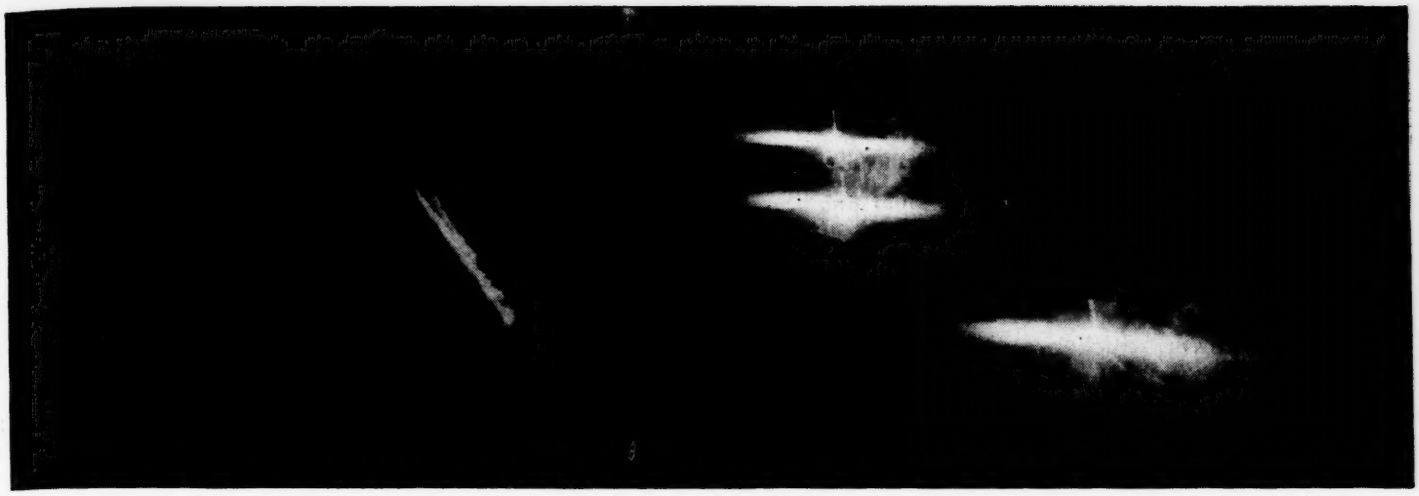

Fig. 2. Removing the rock from tunnel after a blasting.

ditions to a reasonable level and also materially to reduce dust concentration during drilling.

These fans moved a large quantity of air, approximately 10,000 cfm. They were usually mounted in a 3 ft length of piping, sometimes attached to duct work leading directly to the face, and sometimes opening freely at each end with a ½-in. mesh screen guard over the pipe ends. Added resistance of the duct-work reduced the effective fan capacity. As a result, the advantage of direct pick-up in the dust and fog-producing area was nullified, and no measurable difference was found between the two methods. An attempt was made to improve conditions by leading the discharge from this auxiliary fan to the intake of the exhaust main piping, but to no material avail.

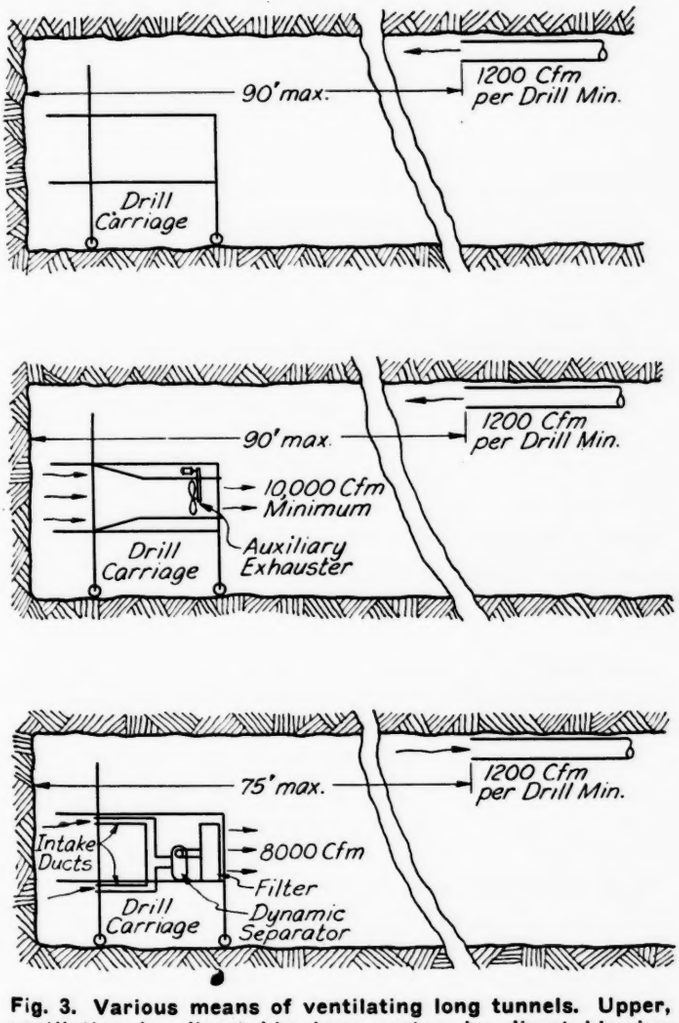

Fig. 3. Various means of ventilating long tunnels. Upper, ventilation by direct blowing; center, by direct blowing plus auxiliary exhaust; lower, direct exhaust plus dynamic separator.

Air measurements were made to determine the cause of fog and dust suppression resulting from the use of this equipment. It was found that there was a considerable condensation of mud on the discharge screen. Discharge temperatures were slightly higher than intake temperatures with a resultant lowering of humidity, but results were not sufficiently definite to allow specific conclusions. It was believed that use of a finer screen might even further reduce fog and otherwise emphasize the observed effects. Repeated tests using from 1 to 3 layers of 10 mesh copper screening; however, showed no measurable difference.

One series of installations involved the use of wet-type, dynamic precipitators placed on the drill carriage but these were not found to have sufficient effectiveness to warrant use of this expensive method of air movement. Dust concentrations in the working zone with this type of equipment recirculating and without any type of after-filter, were found to vary between 20 and 50 million particles per cubic foot of air. When air from these dynamic precipitators was recirculated through paper after-filters, the filters were found to plug very rapidly and to become full of holes. As a result of this they were thought to be more of a nuisance than an advantage. In the "gassy" installations, the discharge from these fans (dynamic precipitators) was led directly back to the exhaust pipe and dust concentrations were found to be approximately proportionate to the air quantity moved. In no case were these as effective in reducing fog density as the high volume propeller fans.

Where portable air moving equipment was necessary to diffuse a temporary high concentration of contamination, venturi type air movers were found to be very effective. These could be carried to the location, attached to one of the ever-present compressed air outlets and run as long as necessary.

Next month Mr. Harris discusses tunnel ventilation equipment.

Packaged Steam Generators

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This article deals only with the packaged, Scotch Marine type steam generators of from 15 to 500 boiler horsepower which are shipped with burner and controls completely assembled.

THE packaged steam generator is a natural outgrowth of the application of automatic firing to conventional boilers of 15 to 500 boiler horsepower which are primarily designed for burning coal in the production of steam for heating and processes other than power generation. Engineers have been applying automatic firing equipment to existing boilers in this class for many years. They have been constantly striving to increase the overall operating efficiency of their equipment and to obtain better control of their steam generating systems. Their objective has been a steam generating system able to adjust itself to the varying load demands of the plant or process. Another factor influencing the application of automatic firing equipment has been the reduction in necessary manpower in the boiler room. Out of the application of automatic firing equipment grew the first packaged steam generator which appeared approximately 15 years ago and included boiler, burner, and controls in a single unit as a step forward to the solution of the problem of steam supply flexibility.

During World War II, several manufacturers became interested in packaged steam generators, and with the interest shown by a larger group of manufacturers, more progress has been made in the packaged steam generator field during the past five years than was made during the ten years preceding. Because much of the development work was done by those interested primarily in oil burning equipment, the available generators are predominantly oil-fired, although some are available also as gas-fired units. The result is that today there is available to the plant operator or consumer a sufficient variety of types and sizes of packaged steam generators to meet a wide variety of application problems.

It is true that there have been many boilers available for every type of steamload, and there have been oil burners available also in a variety of sizes and types to match this demand. However, the matching of the two in a single, semi-portable sales unit was not accomplished until packaged steam generators became available. Formerly, the commercial user of steam, for uses other than power, had no choice but to install a conventional boiler and use hand firing methods, or if he felt so inclined, he could install an oil or gas burner in an attempt to make his installation automatic. There are many installations of this type which are apparently doing a good job.

Design Characteristics

The designers of packaged steam generators have established their design specifications to develop a closely controllable, highly efficient, flexible, compact generator for process steam within the range of 15 to 200 lb per sq in. and 15 to 500 boiler horsepower.

These design considerations are as follows:

(1) Elimination of effect of variables such as draft, location of unit with reference to chimney, and so on;
(2) Close control of the air-fuel ratio;
(3) Arrangement of heating surface to absorb heat from the flame and hot gases in the most efficient manner;

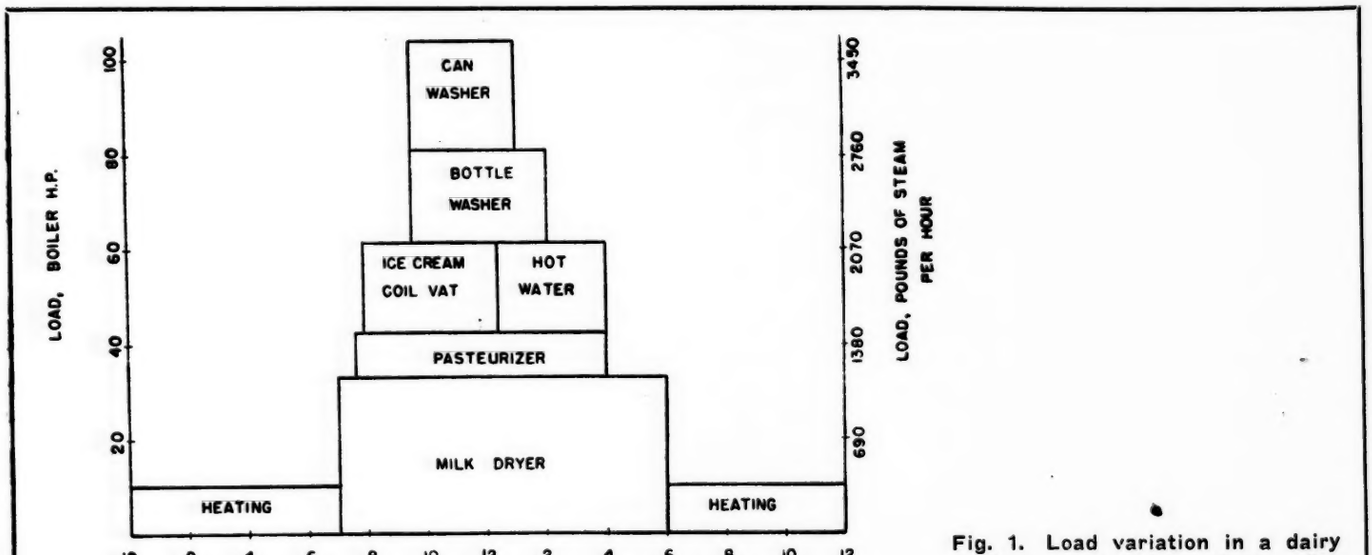


Fig. 1. Load variation in a dairy products processing plant. See example under "Applications."

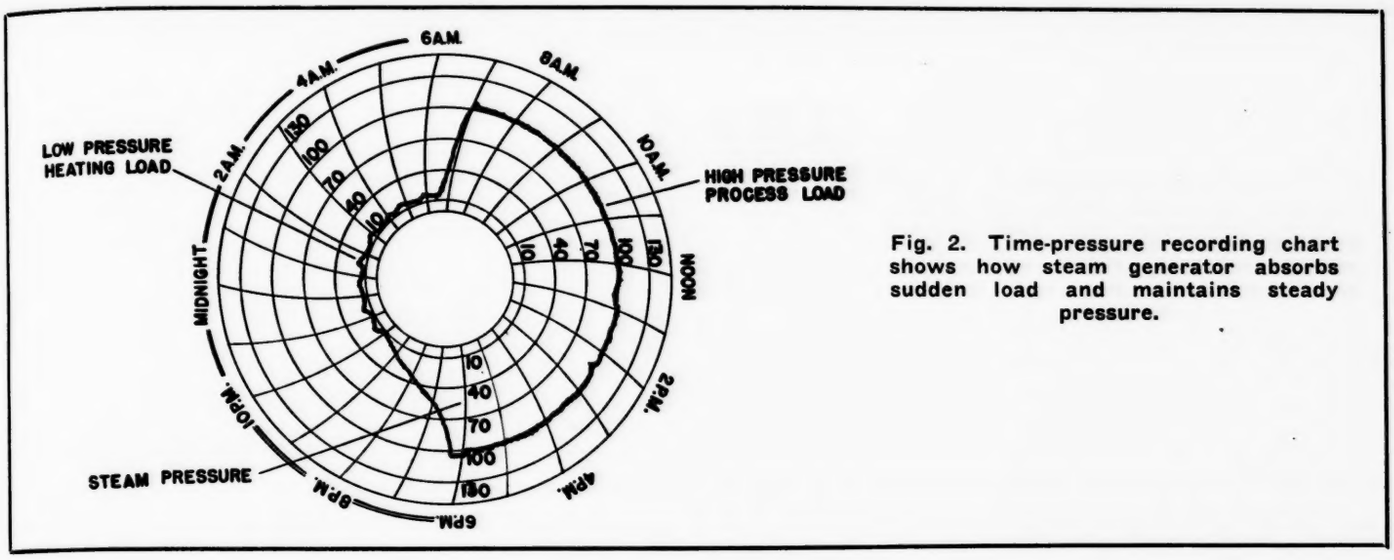


Fig. 2. Time-pressure recording chart shows how steam generator absorbs sudden load and maintains steady pressure.

(4) High overall efficiency;
(5) Refractory held to a minimum;
(6) Quick, easy response to varying load demands;
(7) Unit must be compact;
(8) Unit must be shipped complete to the customer, including burner, controls, wiring, insulation, jacket and chassis;
(9) Each unit must be fire tested and adjusted at the point of manufacture; and
(10) Unit must be able to be used on all types of oils and include the necessary oil preheaters and controls as part of the unit.

With these specifications in mind, there are now packaged steam generators available in sizes from 15 to 500 boiler horsepower actual maximum rating and in design pressures from 15 to 200 lb per sq in. for light or heavy oils to meet every type of commercial process or heating requirement in accordance with present-day demand for efficiency, flexibility, and economy.

Rating

The method used in rating packaged steam generators as compared to conventional boilers needs to be fully understood and respected. Conventional boilers of this class are usually rated in "Rated Boiler Horsepower." This boiler rating is based entirely on the amount of heating surface (1 boiler horsepower for every 10 sq ft heating surface) and has little to do with actual steaming capacity. Packaged steam generator ratings are based on actual steaming capacity (1 boiler horsepower for every 34.5 lb water evaporated per hr from and at 212F), and the packaged steam generator manufacturers' branch of the American Boiler and Affiliated Industries has been discussing the advisability of rating their equipment in pounds of steam per hour.

Boiler ratings for conventional equipment are capable of large overloads. Many such boilers are capable of operating at 200% of rated load, whereas packaged steam generators are designed to deliver their rated output. Thus, in applications where extreme load fluctuations exist, it may be necessary to install multiple units, so that the units may automatically go on or drop off the line, depending on the load. Usually this method may be used without penalizing cost or space requirements. However, even if the initial cost is greater, the higher overall operating efficiencies tend to offset this difference.

Packaged steam generators are capable of operating and maintaining high overall efficiencies; the heating surfaces are matched with the flame shape, thus taking full advantage of the radiant heat emitted from the flame. The gas passes also are arranged to pick up the maximum amount of heat from the hot gases. Standby losses are reduced to a minimum. Since the units do not rely on natural draft, the down draft principle may be used to form a heat trap, thus preventing cold air from being circulated by stack effect through the boiler during standby periods.

Units are of Scotch marine design which allows for internal firing, thus eliminating losses through combustion chamber brickwork.

Each unit built is tested under fire before shipment and all adjustments made to insure maximum combustion efficiency, instead of relying on individual service men in the field to coordinate the many elements.

To be considered by engineers when applying packaged steam generators is the fact that the majority of auxiliaries normally found in the boiler room are a part of the packaged steam generator. An erroneous impression sometimes exists that maintenance is high on a packaged steam generator. As a matter of fact, their maintenance is likely to be low because these auxiliaries are engineered to the unit rather than selected individually. A distinct advantage results from the centralized single responsibility for proper operation of both boiler and burner. Where responsibility of boiler operation, burner operation, piping, and electrical wiring of controls rests with separate organizations, delays are probable during installation and service.

With packaged steam generators, the ease of installation is a feature which should not be underestimated. The equipment can be considered as any other movable piece of equipment. Today, economy of space and flexibility of arrangement and rearrangement are of primary importance, and this feature is invaluable. Because packaged steam generators are received with

all auxiliaries mounted in their proper location on the unit, piping connections and power source are all that are needed.

Since the amount of refractory in packaged steam generators is held to an absolute minimum, the maintenance cost of this refractory should be low. In addition to the saving in initial cost of refractory, the costly shut-down time for its replacement is greatly reduced.

A point of interest is that where boilers rely on natural draft for a large percentage of the air required for combustion, even though draft regulators are employed, it is difficult to deliver the exact amount of air required. Also, when operating with natural draft, it is difficult to control air infiltration. Inefficient combustion and higher fuel costs result when excess air is uncontrolled.

Although in certain localities relatively large stacks are required by local codes, the packaged steam generator actually needs only stack enough to carry the products of combustion beyond the building. Therefore, in many installations, the cost of erecting a suitable stack may be greatly reduced.

The quick response of the packaged steam generator to changes in load is a distinct advantage in many processes. There is a time delay required to heat up or cool off refractory, which in the packaged steam generator is held to a minimum.

Applications

Although this article cannot cover applications of packaged steam generators in a comprehensive manner, the following description of an application in a dairy is presented as a typical example.

At the time the installation was made, the steam generating plant consisted of a conventional, 100 hp stoker-fired boiler. A boiler room attendant was employed to operate the equipment. The operating pressure for the dairy processes was maintained at 100 lb per sq in. and steam at reduced pressure was used for heating.

The packaged steam generator selected was a 100-boiler horsepower (actual rating) unit arranged for full, automatic, modulating firing with No. 6 oil. The unit was equipped with both a high pressure control and a low pressure control, interconnected through a selector switch, enabling the unit to be operated on either low pressure for heating or high pressure for process steam generation.

Fig. 1 graphically shows the steam load requirements in both actual boiler horsepower and pounds of steam per hour against time. Reference to Fig. 1 shows that the unit was used for heating on low pressure operation from 6 p.m. to 7 a.m. Between 7 a.m. and 6 p.m., the load is one of high pressure process steam. This process steam load increases rather rapidly from 7 a.m. to the peak load at 9:30 a.m. The peak load is required until 1 p.m., when it begins to fall off gradually until 6 p.m., when the load is again reduced to that of low pressure heating.

The ability of the packaged steam generator to follow this changing load is illustrated by Fig. 2, which is a standard pressure recording chart. Reference to Fig. 2 shows that the unit maintained a practically constant 100 lb per sq in. operating pressure between 7 a.m. and 6 p.m. This is the same period during which the load was rapidly changing as shown on Fig. 1.

Fig. 3 is an actual photograph of the stoker fired boiler and the modern packaged steam generator which replaced it.

Concerning the economy of operation, actual cost comparison figures show a definite saving in favor of the packaged steam generator. The average fuel cost during 1945, with bituminous coal at $6.65 a ton, for the stoker fired boiler was $7.50 a day. During the early part of 1945, with bituminous coal at $7.00 a

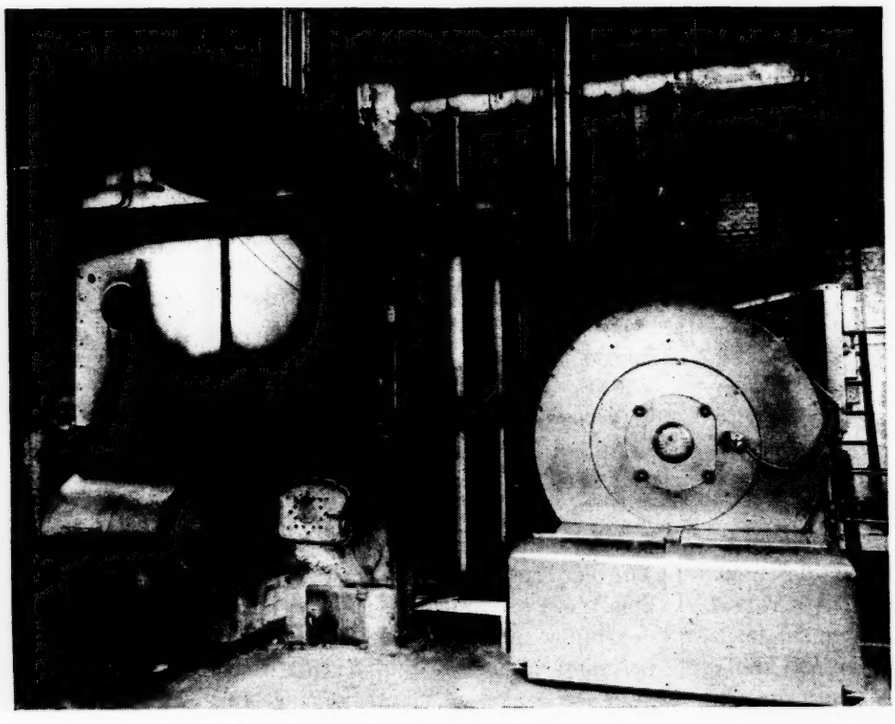


Fig. 3. A stoker fired boiler and the steam generator which replaced it in the example cited above.

How to Use Chart, Fig. 4.

13,750 Btu-per-lb coal costing $8.25 per ton is being used to fire a conventional boiler. No. 6 fuel oil is available at 5c per gallon. To find the cost of No. 6 oil required to equal the steam output from one ton of 13,750 Btu-per-lb coal, with a steam generator replacing this conventional boiler:

First: locate the Btu value of the coal (13,750) on the vertical scale. Draw a horizontal line from this point to the intersection of this line with the line representing the grade and cost of oil (No. 6 oil at 5c).

Second: from the point of intersection, draw a vertical line to the horizontal scale and read the equivalent cost of oil ($6.15).

Third: the approximate saving in fuel cost will then be cost of coal per ton ($8.25) less equivalent cost of oil ($6.15)—or $2.10.

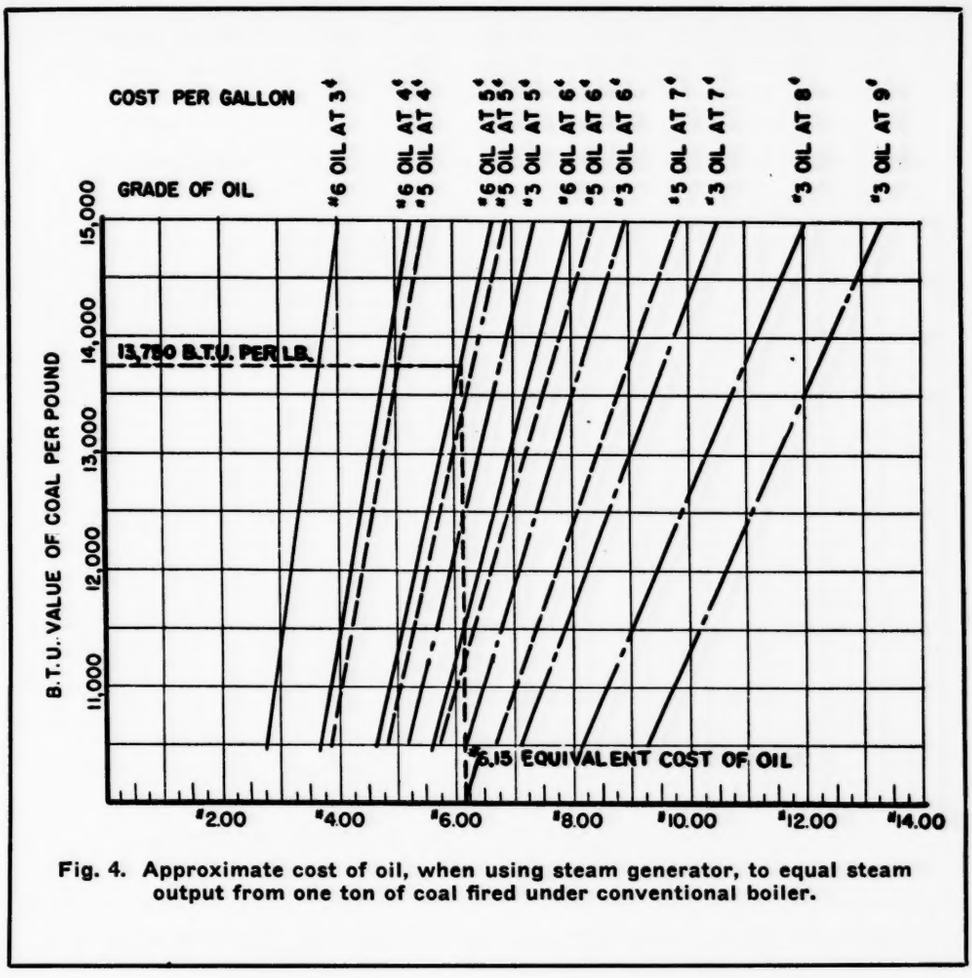


Fig. 4. Approximate cost of oil, when using steam generator, to equal steam output from one ton of coal fired under conventional boiler.

ton, the cost for the same boiler was $8.73 a day. During the latter part of 1945, with No. 6 oil at six cents a gallon, the average cost was $6.41 a day. In addition to this saving in fuel cost, the boiler room attendant was diverted into other work, since his services were no longer required full time. This saving, combined with the elimination of the cinder removal cost, represented a total saving of $2000 a year. During this period, dried milk production was increased 31%.

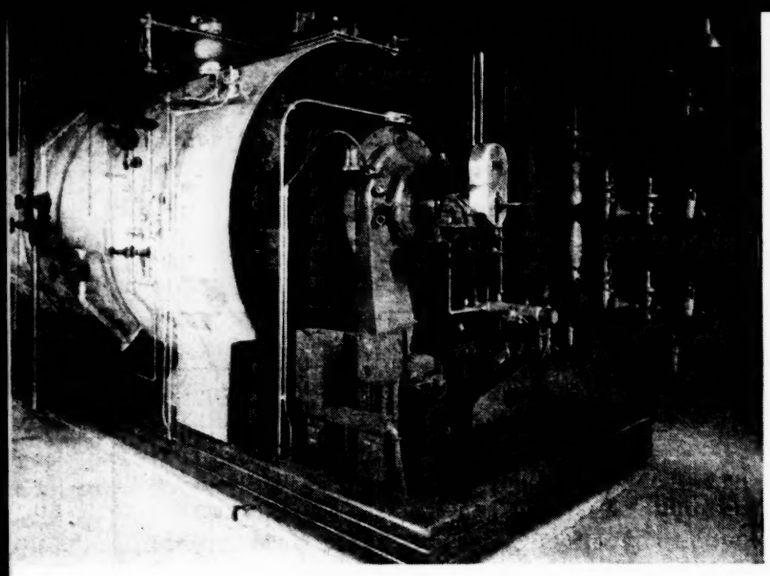
The application considerations of packaged steam generators are essentially the same as those required by conventional boilers. The steam distribution systems must be properly sized and installed to produce the desired results. The selection of the proper fuel oil is a very important consideration and must be carefully analyzed in the same manner as when applying oil burners to conventional boilers. The use of No. 6 oil will show a definite reduction in cost per gallon as compared with the lighter fuel oils. However, this cost saving must be balanced against other important considerations. A few of these are the amount of oil consumed, the availability of this heavy oil, the cost of maintaining this oil at the required temperature for proper handling and burning, and the type of operation of the steam generating equipment, whether continuous or intermittent. Generally, those applications requiring a considerable amount of shutdown time are not satisfactory applications for the use of No. 6 oil.

Fig. 4 is a chart of value in comparing the cost of various grades of oils with coal of different heating values. This chart may be used to advantage in the selection of the proper grade of fuel oil.

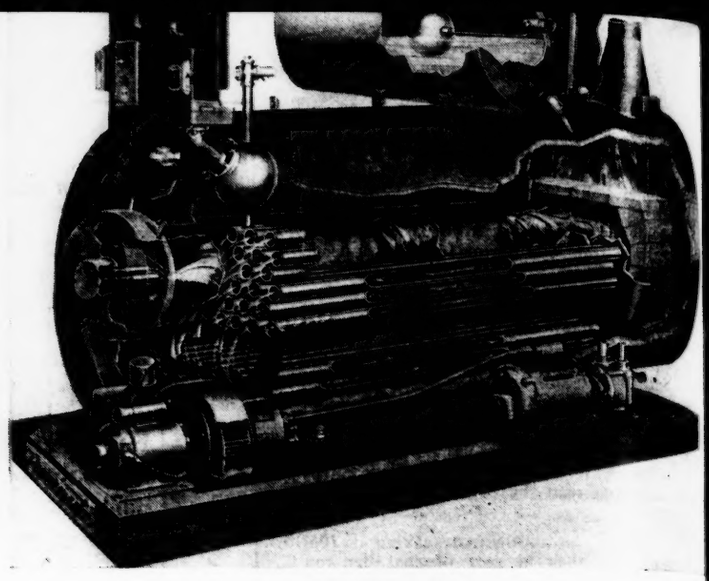
A good rule to use when selecting the fuel to be burned is to determine, from the estimated fuel consumption, if the savings resulting from burning the cheaper grades of oil will pay for the increased cost of equipment in not more than three years. However, this rule must be used intelligently for each separate installation.

For applications where load is seasonal, or has wide fluctuations, multiple installations will effect considerable economies. Under these conditions, it is possible to operate a portion of the installed units under the lighter load requirements and all of the units under the heavy load requirements. In certain textile mill applications, the process load may be only ten or fifteen percent of the total load. Under such conditions, it is more efficient to operate several packaged steam generators, rather than one large unit. The smaller units may be located near the process load, thus reducing piping losses which exist in steam distribution from a central boiler plant.

Packaged steam generators are being applied daily to high pressure process steam loads in dairies, laundries, dry cleaning establishments, canneries, and other processing plants. Low pressure units are being applied to heating loads, bottling plants, and industrial uses requiring steam for a variety of purposes including emergency stand-by for district heating services.



Cleaver-Brooks



Orr & Sembower

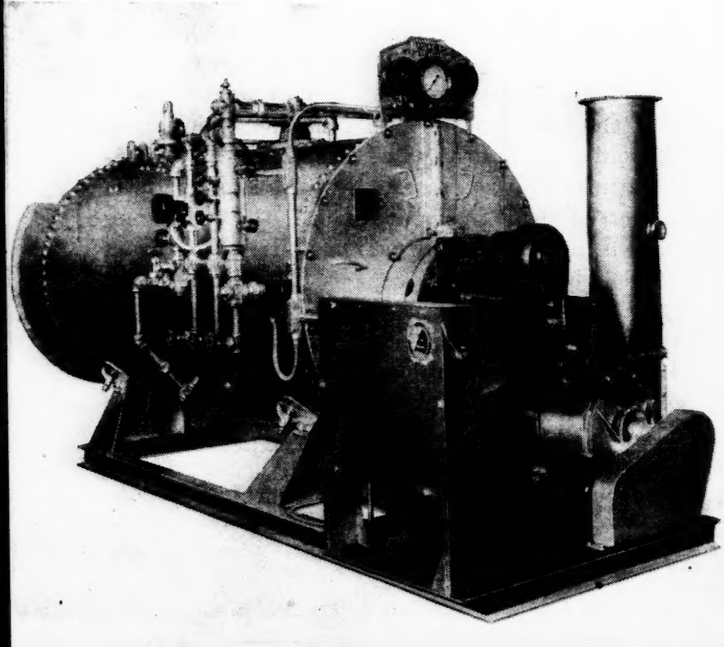
Representative Packaged Steam Generators

The photographs on this page show four typical models of the packaged steam generators discussed in the previous pages. Table below is a recap of capacity ranges and physical sizes of these and other makes.

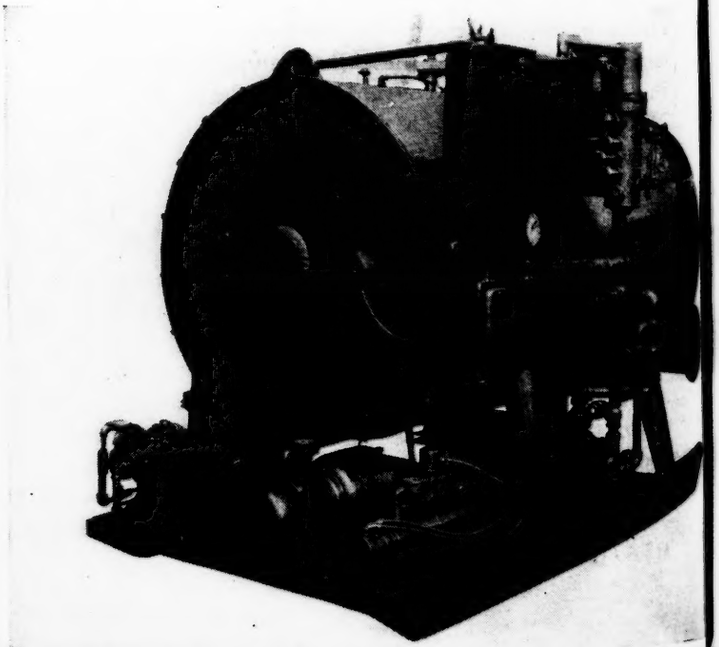
Make	Fuel	Capacity Range			Overall Dimensions, In.		
		Boiler[1] Horse-Power	Pounds Steam per Hour from and at 212F	Pounds per Square Inch	L	W	H
Ames Iron Works	Oil and/or gas	10-300	350-10,500	15-200	54-224	26-84	34-96
Cleaver-Brooks	Oil, No. 3, 5, 6	15-500	525-17,000	15-200	138-266	70-143	58-121
Cyclotherm	Oil and/or gas	5-200	175-6,900	15-200	54-224	26-84	34-96
Johnston Bros.	Oil, heavy and light	60-500	2100-17,000	100-200	176-328	81-124	79-129
Orr & Sembower	Oil, No. 2, 3, 4, 5, and 6	15-156	525-5,250	15-150	74-214	41-68	66-116
Preferred Utilities	Oil, heavy and light[2]	20-500	700-17,000	15-200	107½-323	64¾-175⅛	60-125
Vapor-Clarkson[3]	Oil, heavy		300-3,000				
York-Shipley	Oil, No. 3, 5, and 6	15-100	525-3,500	*	89½-185	46½-78	60½-106

[1]For EDR steam, multiply boiler horsepower by 139.
For Btu per hour output, multiply boiler horsepower by 33,500.
[2]Combination natural gas and fuel oil burner also available.
[3]See "New Equipment" section beginning page 105, this issue.
*High and low pressure lines made.

Preferred Utilities



York-Shipley



Heat Transfer from Ground to Water in Buried Coils

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Cornell University, Ithaca, New York

A vital factor to consider in designing a heat pump installation is the availability of a heat source at an essentially constant temperature. Inherent disadvantages of using outside air or deep well water as a heat source* have prompted this study of the utilization of the ground as a possible heat source for the heat pump. The primary purpose of the investigation was to determine the expected rate of heat transfer from the ground to cold water which is flowing through a coil buried in the ground. The investigation was conducted under the supervision of Professor C. O. Mackey in the Mechanical Engineering Laboratory at Cornell University, Ithaca, New York.

THE ground coil consisted of 40 feet of ⅜-in. O.D. type L copper tubing bent into five 7-ft lengths spaced 1½ ft apart and buried 5 ft below the surface under the heated West Mechanical Engineering Laboratory of Cornell University, Two 3⅜-in. wrought iron pipe risers 5 ft long were located vertically in the ground as shown in Fig. 1. The coil was situated near the outside wall of the laboratory. The total length of the copper tubing and wrought iron risers was 50 ft and the total outside surface area below ground was 5.35 sq ft.

Ground

The ground in which the coil was buried is a top-set bed of an old delta. The condition of the soil was sandy and dry, containing 0.038 lb of moisture per pound of dry sand. Numerous rocks ranging in size from approximately ½ in. to as large as 7 or 8 in. in diameter were distributed throughout the area. A fairly representative sample of the ground was tested for thermal conductivity, and the value of "k" obtained was 0.31 Btu per ft per hr per sq ft per F at an average ground temperature of 57F. Due to the physical limitations of the conductivity testing apparatus, it was impossible to place large rocks in the sample, and therefore the value of k obtained was probably higher than actual. The soil condition is notably poor for conduction of heat because of the large pore spaces.

Temperature Measurements

Ground Temperatures. Eight copper-constantan thermocouples were located at various depths in the ground as follows: One at the surface, three at 2 ft, three at 4 ft, and one at 7 ft. (See Fig. 1) A Leeds & Northrup potentiometer was used to determine the e.m.f., and the temperatures were read from the standard calibration curves.

Water Temperatures. Since the temperature difference of the water entering and leaving the ground coil was small, a very accurate determination of this difference was required. To accomplish this a 4-couple thermopile was constructed and placed in the inlet and

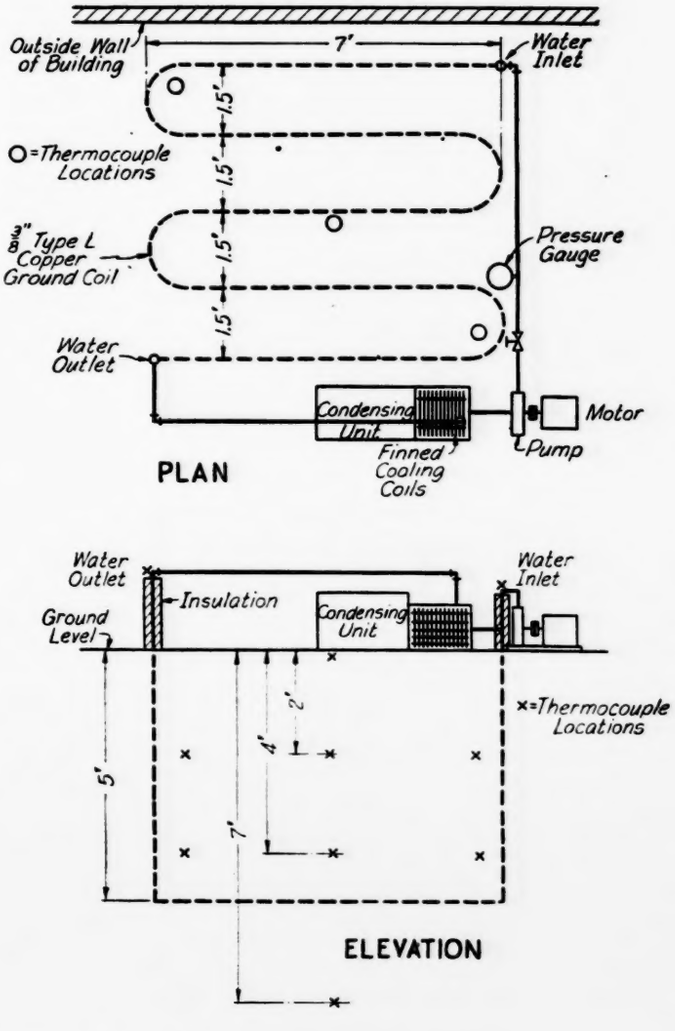


Fig. 1. Layout of test apparatus.

*Kemler, E. M., in HEATING AND VENTILATING, Dec. 1946, p. 75.

outlet connections. Swirl strips were located at the upstream side of each set of couples to insure thorough mixing of the water. By suitable electrical wiring it was possible to measure the inlet water temperature as well as the temperature difference between inlet and outlet connections. Since measurements of inlet and outlet water temperatures were made at points above the ground surface, insulation was placed around these sections of pipe to reduce any heat gain from the laboratory building. The maximum calculated heat gain from the air was of the order of 20 Btu per hr for this small length of above-surface pipe.

To obtain a sufficiently low inlet water temperature, a ½-hp Westinghouse Freon-12 condensing unit was used in conjunction with a finned evaporator coil. This coil was placed in a tank through which the water circulated, a small agitator being used to prevent stagnation.

The water was circulated through the ground coil by a Westco centrifugal pump driven by a ¾-hp motor. A uniform flow rate was obtained by adjusting a gate valve on the pump discharge so that a pressure gage after the valve read a constant value. Initially the rate of flow was determined at this pressure indication by a weigh tank and stop watch.

Procedure

The test was conducted from 10 a.m., March 24, 1947, until 4 p.m., March 27, 1947, a total of 78 hours. Measurements of the ground temperature, entering water temperature, and temperature difference were taken approximately every two hours. Outside air temperatures were obtained from the weather station of the New York State College of Agriculture at Cornell. The pressure gauge was maintained at a predetermined value of 50 lb per sq in., corresponding to a flow rate of 940 lb per hr ($\pm$ 20 lb per hr).

Results

On Fig. 2 is plotted the heat transferred to the water, Q, against the time. The value of Q was determined by multiplying the rate of flow of water (in pounds per hour) times the temperature difference of the water. It can readily be seen that the value of Q is 1600 Btu per hr at the beginning of the test, but decreases until, after 40 hours have elapsed, the value has essentially levelled off at approximately 700 Btu per hour.

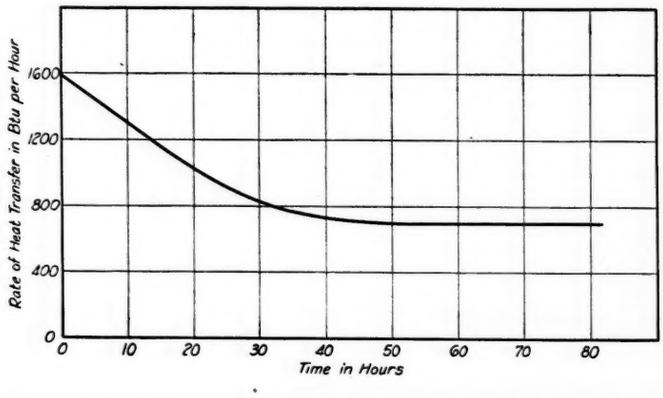


Fig. 2. Heat transfer vs. time. Ground coil data; 40 ft of ⅜-in. type "L" copper tubing; 10 ft of ⅜-in. wrought iron pipe; total outside area, 5.35 sq ft; buried at depth of 5 ft; average undisturbed ground temperature, 57F; average inlet water temperature, 38F.

This levelling off may be explained as follows: Initially the ground temperature immediately surrounding the coil is a certain value, and as cold water flows through the coil, heat passes from this ground to the water, and thus the temperature of this ground decreases. As this ground temperature decreases, heat from the surrounding ground follows the temperature gradient to the ground adjacent to the coil and then to the water. Apparently after a period of time, a state of equilibrium is reached where the heat leaving the ground immediately adjacent to the coil is replaced by an equal amount of heat from the surrounding ground.

It should be realized that the value of 700 Btu per hr at which the curve levels off is not all due to heat transfer from the ground. As previously shown, there is a gain of 20 Btu per hr from the pipe above ground. In addition, there is a gain of heat due to the friction of water flowing through the pipe. This amounts to approximately 80 Btu per hr. (See Appendix for calculations.) These two gains reduce the heat actually transferred from the ground to a value of 600 Btu per hr, rather than 700. The value of 600 Btu per hr is used in calculating the values shown in the conclusion.

In Fig. 3, temperature versus time, certain trends are indicated. The variation of the outside air temperature affected the inside air temperature and hence the ground surface temperature. The trend at a 4-ft depth was a decrease in temperature. Thus the outside air temperature has not caused a fluctuation in the ground temperature at 4 ft. In a further investigation it would be interesting to determine the variation in ground temperature at increasing distances from the ground coil.

Conclusions

The equilibrium rate of heat transfer from ground to water in the coil was found to be 6.24 Btu per hr for each square foot of coil surface and for each degree of temperature difference between *undisturbed ground* at coil level and average water temperature. The Rudenberg equation for a long cylinder with an outside diameter of ⅜ in. buried in a semi-infinite solid with K = .31 Btu per ft per hr per sq ft per F at a distance of 5 ft below the surface predicts a value for the rate of steady heat conduction of 2.4 Btu per hr per sq ft per F. (See Appendix for Calculations.)

The observed product for the thermal resistance of the ground and the thermal conductivity of the soil was 0.0496 ft. In other words, the insulating effect of all the ground that surrounds the coil is equivalent to a solid layer of 4.31 inches thickness. Since wet clay may have a thermal conductivity of at least four times that of the dry sand and gravel in which the test coil was imbedded, it would not be surprising to find overall rates of heat transfer as high as 25 Btu per hr per

sq ft per F for coils imbedded in wet clay. If the temperature of a refrigerant in a buried direct expansion coil were low enough to freeze ice in damp soil in the vicinity of the coil, heat transfer rates as high as with clay might be obtained because the thermal conductivity of ice is about four times that of the soil in this test.

Although the test coil was buried under a heated laboratory near an outside wall, the authors believe that heat transfer rates of nearly the same magnitude would have been obtained if the coil had been at the same depth under the surface of ground outside of the building wall. They appreciate the need of further study to substantiate this guess.

This test, along with other tests conducted by the authors, indicates that short-time, intermittent rates of heat transfer of about 2.5 times the equilibrium rate might be obtained.

The authors believe that an extended surface coil would produce better heat transfer, due to the greater area of contact, and they recommend further experiments with this type of coil buried in the ground.

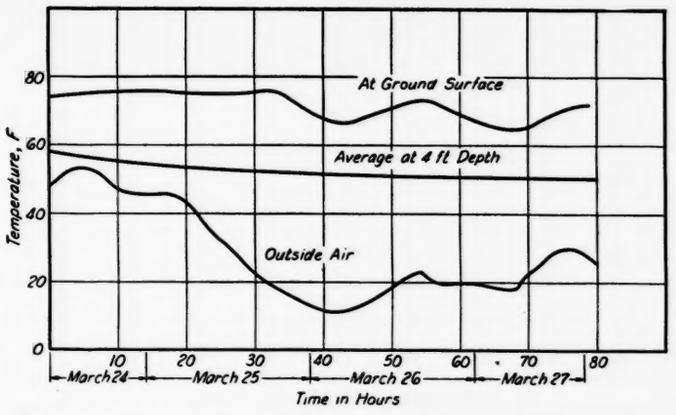


Fig. 3. Temperature vs. time for outside air, ground surface, and ground at 4-ft depth.

The ground appears to be an attractive source of heat for the application of a heat pump in northern sections of this country. The necessary length of a buried coil does not appear to be prohibitive.

Appendix

Heat gain from air to above-surface pipe:

Length of above-surface pipe = 4.5 ft of 3/8-in. pipe with 1 in. of insulation.

$$\text{Surface area} = 4.5 \times \frac{2.375\,\pi}{12} = 2.80 \text{ sq ft}$$

Air temperature = 75F Water temperature = 37F

$$\text{Insulation resistance} = \frac{r_2}{k}\log_e\frac{r_2}{r_1} = \frac{1.1875}{0.5}\log_e\frac{1.1875}{0.1875}$$

$$= 4.37 \text{ hr per sq ft per F per Btu}$$

Combined conduction and radiation coefficient = 1.605

Neglect thermal resistance of tubing and water film.

$$\text{Total thermal resistance} = 4.37 + \frac{1}{1.605} = 4.99$$

hr per sq ft per F per Btu

$$\text{Heat transfer, } q = \frac{2.80\,(75-37)}{4.99} = 20.5 \text{ Btu per hr}$$

Heating effect of water friction in coil:

$$f = \frac{0.2}{(N_r)^{0.2}} \qquad N_r = \frac{0.315\,(8)\,(62.3)}{12\,(1.6)\,(0.000672)} = 12{,}180$$

$$f = \frac{0.2}{6.55} = 0.03$$

Assume the equivalent frictional length of pipe to allow for bends and pipe fittings to be 60 ft.

$$h_f = f\,\frac{L}{D}\,\frac{V^2}{2g} = 0.03\left(\frac{60\times 12}{0.315}\right)\left(\frac{8\times 8}{64.4}\right)$$

= 68.5 ft of water in test section.

$$\text{Heating effect} = \frac{68.5\times 940}{778} = 82.7 \text{ Btu per hr}$$

The combined effect of heating due to friction and gain of heat between sections of temperature measurement and ground is about 100 Btu per hr.

Thermal resistance of the ground:

Assume an equilibrium rate of heat transfer during test to be (700 — 100) or 600 Btu per hr.

Average ground temperature = 57F

Average water temperature = 39F

$$\frac{1}{U_2} = \frac{r_2}{k_w r_1} + \frac{r_2}{k_t}\log_e\frac{r_2}{r_1} + R_g$$

$$U_2 = \frac{600}{5.35\,(57-39)} = 6.24 \text{ Btu per hr per sq ft per F}$$

Based on outside surface of tube, the total thermal resistance is $\frac{1}{6.24} = 0.1605$ hr per sq ft per F per Btu

Thermal resistance of copper tube is

$$\frac{0.375}{24\ (224)}\log_e\left(\frac{0.375}{0.315}\right) = 0.000012$$

From McAdams' "Heat Transmission,"

$$h_w = \frac{150\,(1.429)\,(8)^{0.8}}{(0.315)^{0.2}} = 1420$$

Thermal resistance of the water is

$$\frac{0.375}{0.315\ (1420)} = 0.00084$$

Thermal resistance of ground, R_g is 0.160 hr per sq ft per F per Btu

Using $k_g = 0.31$ Btu per hr per sq ft per F, $R_g k_g$ = 0.0496 ft.

Comparison with Rudenberg equation:

$$R_g k_g = \frac{D_2}{2} \log_e \frac{2L}{D_2} \left\{ 1 \Big/ \frac{\log_e \frac{L}{2Z}}{\log_e \frac{2L}{D_2}} \right\}$$

$$= \frac{0.375}{2 \times 12} \log_e \frac{2 \times 50 \times 12}{0.375} \left\{ 1 \Big/ \frac{\log_e \frac{50}{2 \times 5}}{\log_e \frac{2 \times 50 \times 12}{0.375}} \right\}$$

$= 0.129$ ft.

$$\text{Conductance} = \frac{0.0496}{0.129} \times 6.24 = 2.4 \text{ Btu per hr per sq ft per F}$$

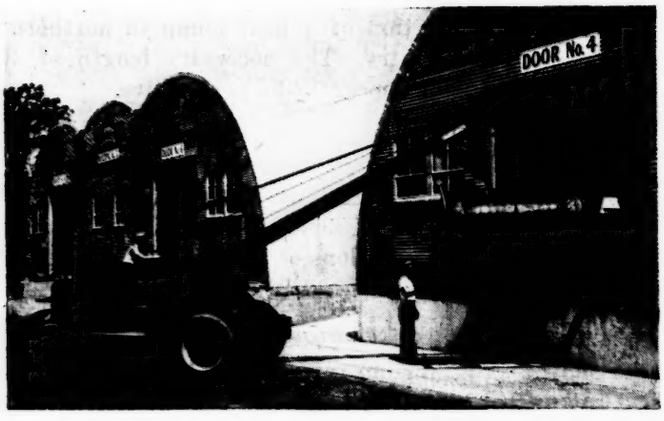
Calculation of equivalent insulation thickness of ground:

$$0.160 = \frac{r_1}{k} \log_e \frac{r_2}{r_1} = \frac{0.375}{2 \times 0.31 \times 12} \log_e \frac{2r_2}{0.375};$$

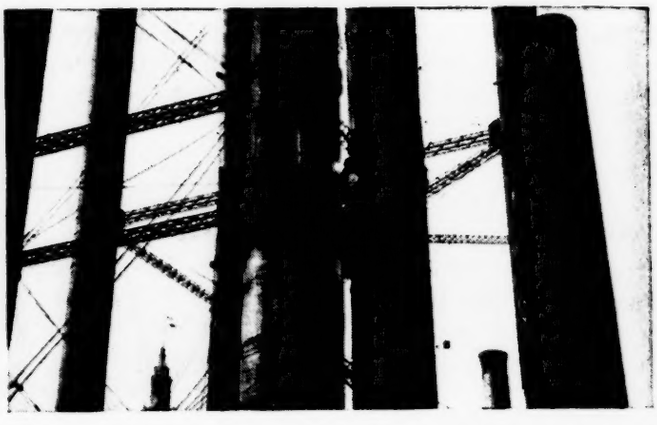
$$\text{from which } r_2 = \frac{24 \times 0.375}{2} = 4.50 \text{ in.}$$

$$\text{Thickness} = 4.50 - 0.1875 = 4.31 \text{ in.}$$

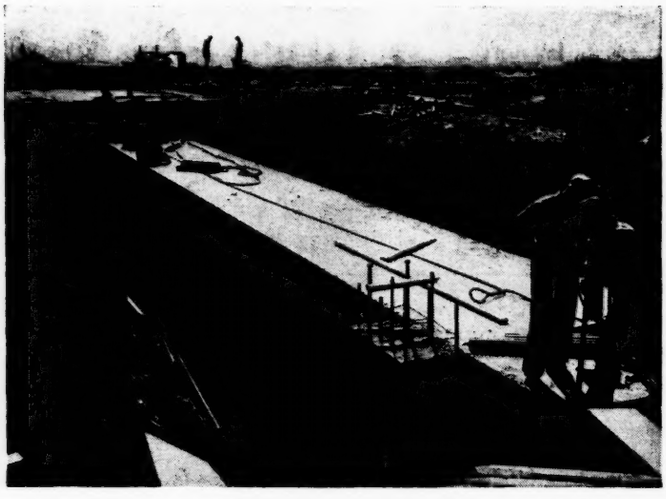
PICTURE PARAGRAPHS



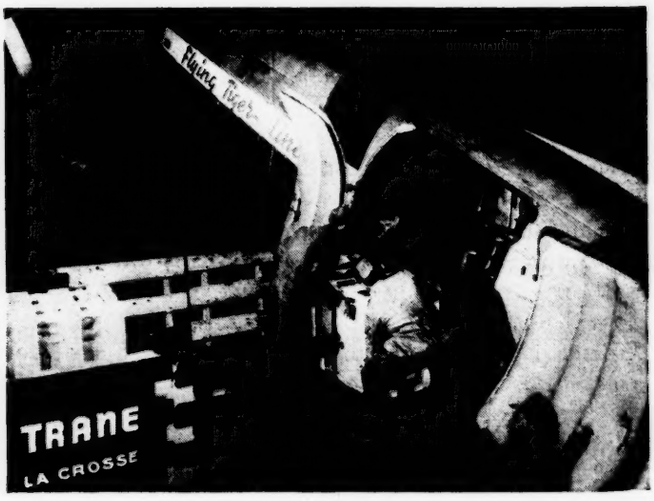
Roustabout crane, said to be the largest of its type in use in central Pennsylvania, is gasoline driven and has a lifting capacity of six tons. It is being used for heavy loading and unloading work at the hugh new Quonset plants of York-Shipley, where a complete line of "packaged", oil-fired heating equipment is being mass-produced for the residential market.



Cleveland Electric Illuminating Co.'s smokestack has been reinforced by welding ¼-in. steel plates to the steel shell from the foundation of the stack up to a 60-ft height. The top part of the stack, which is 120 ft high, was similarly reinforced several years ago. Mertz Co. of Cleveland was in charge of the repair and used welding equipment furnished by The Lincoln Electric Co. of Cleveland.



Utility services required in the new Diesel engine factory under construction at Caterpillar Tractor Co., Peoria, Illinois, will be introduced into the factory through a service tunnel connecting with present utility lines situated more than 700 ft from the site of the new building. Tunnel is of reinforced concrete construction of rectangular cross section 7 ft high and 7 ft wide with an overall length of 715 ft. Lines to be run through the tunnel include a 10-in., 150-lb steam line; a 6-in. condensate return line; an 8-in. compressed air line; a 6-in. domestic water line; and a 10-in. mill water line.



A DC3 cargo plane of the famed Flying Tiger line picked up a shipment of 10 Trane Custom-Air conditioning units at the La Crosse Airport for quick shipment to New York. The Flying Tiger plane was chartered by Almirall & Company, Inc., New York City, general contractors for the company where the units were to be used. These 10 units were a portion of an order required in a prominent office building in New York City. The deadline for this installation was set, and a plane was chartered to expedite shipment. The pilot, D. W. Dyke of Flushing, Long Island, brought the units into New York

Utilizing the Flash of Condensate Returns for the Heating of Water

T. W. REYNOLDS
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Abbott, Merkt & Co., Engineers, New York, N. Y.

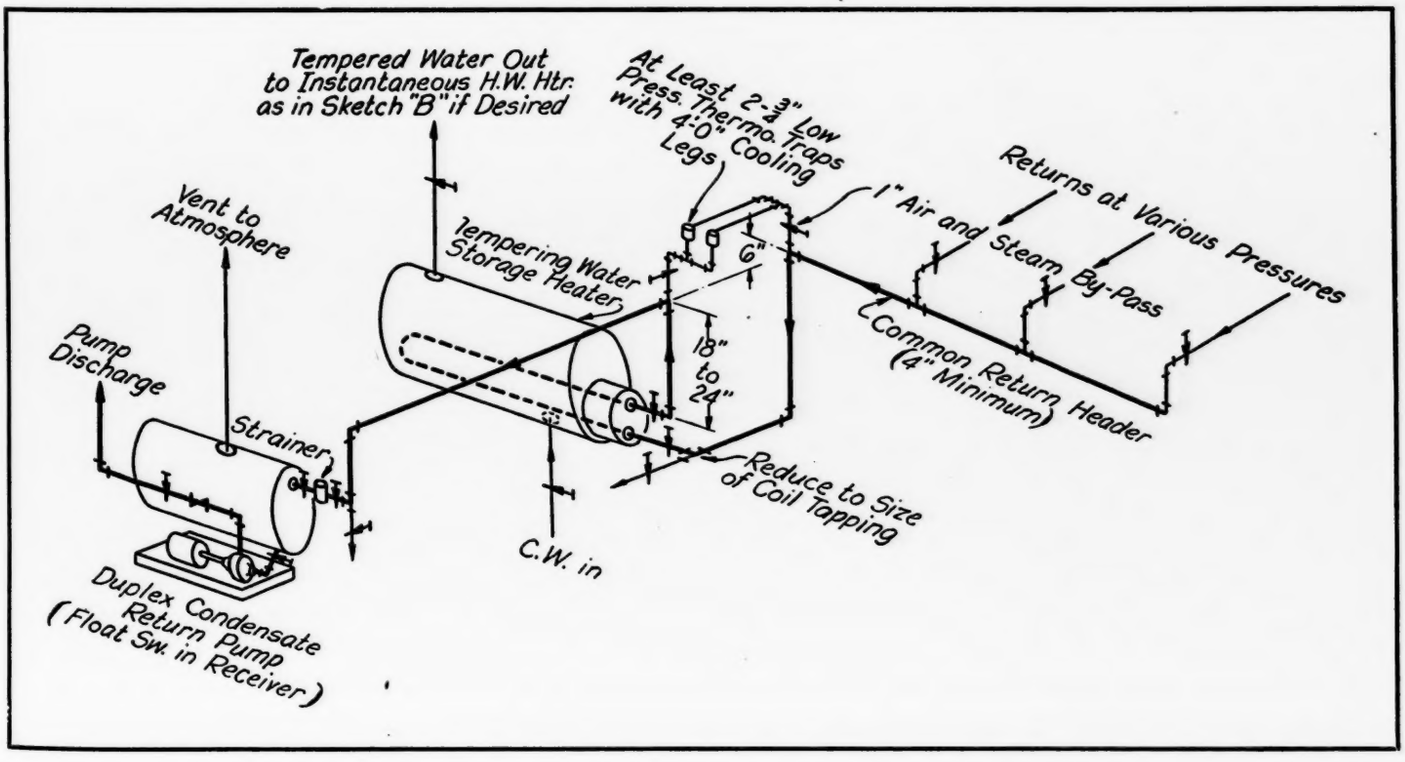
A piping arrangement is given for cooling condensate and using flash steam and excess heat for incidental tempering or full heating of water for both boiler feed and hot water storage supply. A means of including an additional instantaneous water heater to assure domestic hot water service requirements is described.

So much interest was shown in recent articles* on the matter of condensate return that the accompanying sketches "A" and "B" are herein offered as additional thoughts on the subject. The methods as shown are suggested primarily for the cooling of condensate to a temperature low enough for the pumps to handle, but with the incidental advantage that the flash steam of the condensate may be utilized, either to temper water or to fully heat water for a hot water storage tank.

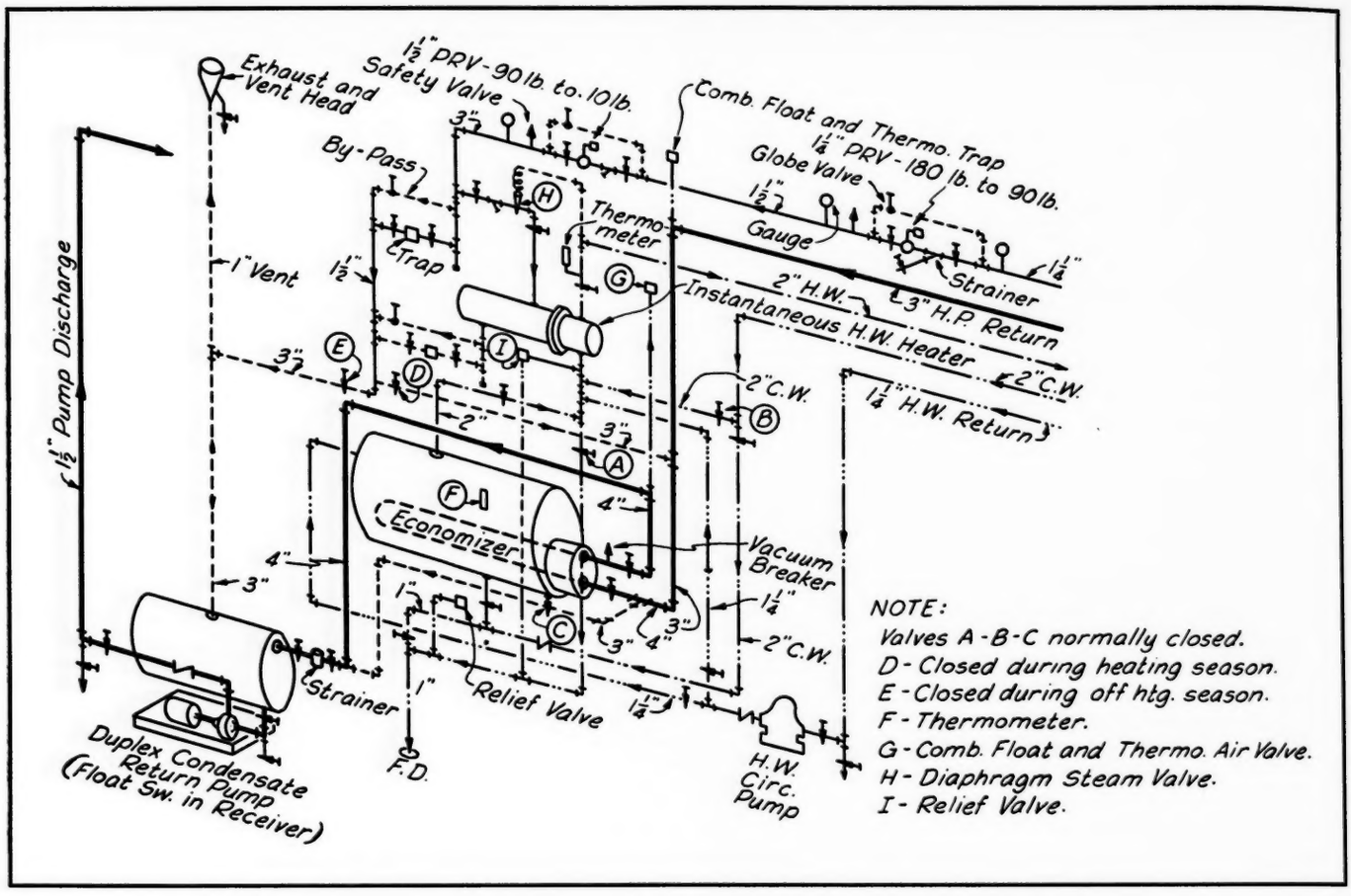
Where, however, the flash can be fully or in the main utilized the year around by the exhaust steam main and for the feed water heater, it is better to flash the steam directly to the exhaust steam main and feed water heater. It is, of course, more important that the boiler feed water be heated to a temperature sufficient to prevent strain on the boilers than it is merely to heat domestic hot water economically. Where the flash can be best used directly as mentioned, the setting of the pressure reducing valve supplying make-up steam to the exhaust steam main should be somewhat lower than the pressure at which the back pressure valve will waste steam to the atmosphere.

Referring to Sketch "A", there is a water seal of from 18 in. to 24 in. and a bottom connection to the heater, both of which tend to keep the coil in the tempering water heater flooded and therefore reduce the temperature of the condensate to as low a point as possible before the condensate passes on to the pump. The coil in the heater should be big enough to pass all of the condensate and condense any steam and also so as to

*See HEATING AND VENTILATING, issues of May, July, September and December of 1946, with particular reference to the last mentioned issue.


Tempered Water Out to Instantaneous H.W. Htr. as in Sketch "B" if Desired
At Least 2-3" Low Press. Thermo. Traps with 4'-0" Cooling Legs
Returns at Various Pressures
1" Air and Steam By-Pass
Vent to Atmosphere
Tempering Water Storage Heater
6"
18" to 24"
Common Return Header (4" Minimum)
Pump Discharge
Strainer
Reduce to Size of Coil Tapping
C.W. in
Duplex Condensate Return Pump (Float Sw. in Receiver)

Sketch A.—Method of reducing condensate temperatures and conserving heat from process returns at various pressures.



Sketch B.—Diagrammatic arrangement for reclamation of heat from process returns.

give sufficient transfer of heat due to the lower temperature differential as the size of coil is increased. The condensate should be passed through the coils. All scale deposits from the tempered water will then form on the outside of the coils where they can be more easily removed.

It will be noticed that the various returns at various pressures are connected into the top of a common return header. This header should be at least 4 in. in size as an aid to the establishment of a common pressure in the header. An air and steam by-pass over the water seal is provided as shown. This by-pass is fitted with a sufficient number of low pressure thermostatic traps as connected by cooling legs at least 4 ft in length.

When only air comes along the traps will open to pass the air along to the pump receiver and its vent, but when steam contacts the traps they will close. The steam is then forced through the heater, providing the common pressure in the header exceeds the equivalent in static pressure of the water seal. In the case shown this is about ¾ lb. The return header should be kept from 4 to 6 in. above the water seal as shown. This will provide a static head sufficient to induce flow, yet leave the header free from water, so that air can be passed along to the traps.

If, due to physical conditions, the return to the pump must enter the pump below its inlet, then an air line must be carried to the pump. If lifts are required at the pump, the pump cannot be run at night merely by float switch operation. There are installations where a vacuum pump is run taking heating and process returns in the heating season with the process returns manually valved off in the off heating season so as to go to a condensate pump. Where, however, the vacuum pumping set is large enough, so that it can be furnished with two pumps for air and two for condensate, then the same pumping set can be operated year around. In such an installation (duplex air and water type), one air and one water pump is operated in the heating season and only one water pump in the off heating season. Thus, no manual change over to other piping is required as the season changes, nor is a second pump for condensate only required.

Sketch "B" is somewhat similar to the arrangement shown by sketch "A". It is taken from an actual installation, given in more detail, and includes the addition of a hot water return circulating pump and also an instantaneous hot water heater, thus insuring the heating of the domestic hot water to its required service temperature. By proper manipulation of valves "D" and "E" as noted in the sketch, the higher pressure drips from the instantaneous heater may be separated in the heating season, so as not to be mingled with the other returns at lower pressure, going instead directly to the pump receiver and there flashing through its vent.

It will be noted that all equipment is fully by-passed, properly relieved of excess water pressure and may be fully drained. Both the cold water and the pumped hot water may go to either heater when and as desired.

Evaluating Performance of the Atmospheric Cooling Tower

JAMES G. DeFLON
Chief Cooling Tower Development Engineer, The Fluor Corp., Ltd.

This is a discussion of the various design factors such as temperature range, wet bulb temperature, tower height and wind velocity that influence the performance of the atmospheric cooling tower. Problems in tower performance analysis are included.

IN atmospheric towers, water is pumped to the top of the tower, where it is discharged through a distributing system. Cross-section of a typical atmospheric cooling tower is shown in Fig. 1. As the water begins its downward flow, it is broken up and redistributed by the decks that comprise the filling of the tower. This continually creates newly exposed cooling surface for air to encounter passing horizontally through the tower. The redistribution insures even concentration of water throughout the tower during its entire fall.

Although initial cost for an atmospheric cooling tower, designed for a 3-mile wind, is about the same as that for a mechanical draft tower, there are certain important limitations governing its performance. The tower must be located broadside to the prevailing wind in an exposed area. Any surrounding structures, hills or other barriers would tend to block the wind.

Originally, the main objection to atmospheric towers was the excessive spray loss during periods of high winds. This loss was caused by the lack of a means for separating entrained water from air in the conventional louver-type tower. Now most manufacturers have solved this problem by incorporating drift eliminators into the louvers.

The method of determining size and performance of an atmospheric type tower is shown on the accompanying charts. They show the method of estimating the size of an atmospheric type cooling tower to meet a given demand. *The cooling capacity of any tower, with a given wet bulb temperature and wind velocity, varies with the water concentration.* Therefore, the problem of calculating tower size becomes nothing more than obtaining the correct water concentration for a tower of chosen height that will operate with a certain wind velocity and wet bulb temperature. Once this water concentration factor is obtained, the area of a given height tower can easily be calculated by dividing the gallons per minute circulated by the concentration factor.

The concentration required to produce desired cooling depends primarily on the following conditions:

(1) Temperature range ($T_1 - T_2$)
(2) Approach to wet bulb temperature ($T_2 - T_{wb}$)
(3) Tower height
(4) Wind velocity
(5) Wet bulb temperature (T_{wb})

Because of the infinite number of possible combinations of those values, it is impractical, if not impossible, to have one curve from which to obtain the correct concentration factor. Fig. 2 gives the required concentration for cooling water through a certain range and with a certain approach to the wet bulb temperature. This curve assumes a wet bulb temperature of 70F, tower height of 35 ft, and a wind velocity of 3 mph. Let us assume that the conditions given are unity. Then should any of these three values change, the concentration would have to be corrected for the

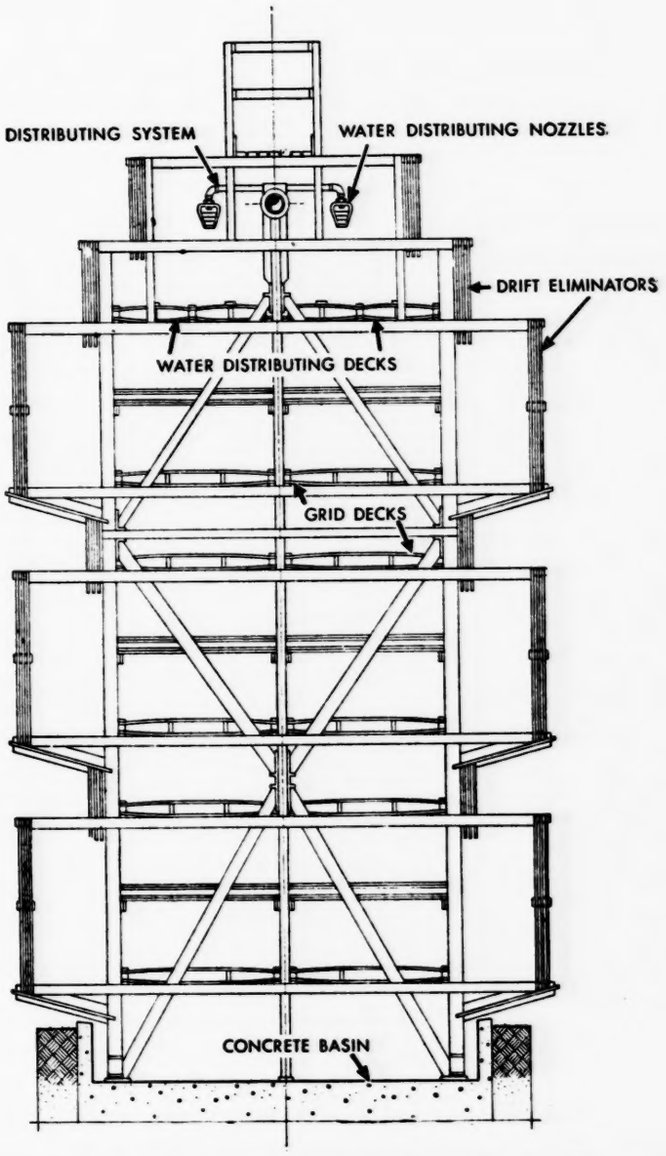


Fig. 1. Section of atmospheric cooling tower.

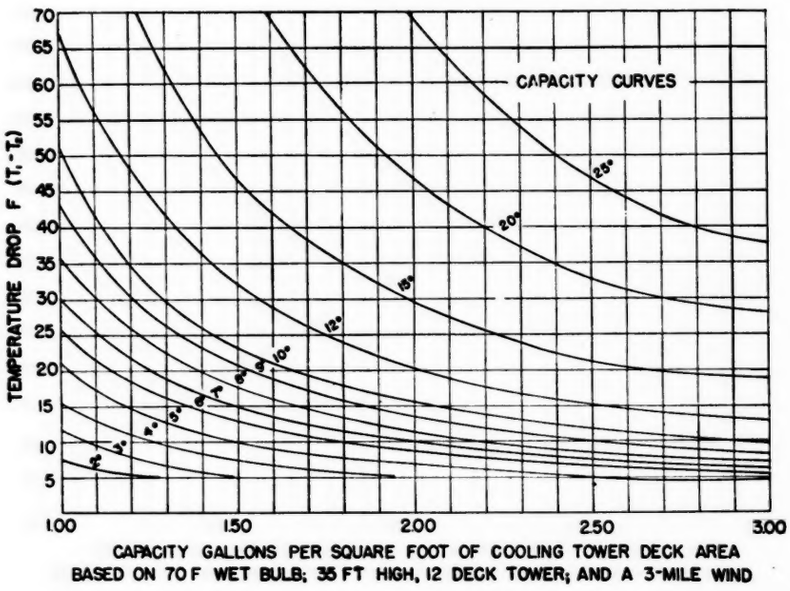


Fig. 2. Capacity curves for atmospheric-type cooling towers.

new conditions by using one or more of the correction factors shown on Fig. 3, 4, and 5.

Let us see how a variance of any of the three conditions mentioned would affect the concentration of water.

(1) *Wind Velocity.* The higher the wind velocity, the greater the amount of air that goes through the tower. This results in greater cooling. Therefore, when the wind velocity is higher, the concentration can be greater and still obtain equal cooling.

In deciding upon a design wind condition, it is well tc bear in mind that there is a period occurring during the day and night in which the prevailing wind shifts. When this occurs, there is a short period when there is little or no air movement. At such a time, the tower water temperatures will rise from 2 to 5F, depending on the duration of the calm and the design wind velocity, and should be evaluated in determining a design wind velocity. However, one should be aware of the fact that many atmospheric towers are designed to successfully operate at a zero wind condition.

(2) *Tower Height.* In general, with atmospheric as well as mechanical draft towers, the greater the cooling range and the closer the approach to the wet bulb temperature, the higher will be the tower required to give sufficient time of contact between water and air to accomplish the desired cooling. In atmospheric towers, the performance is limited by both maximum and minimum water concentrations. Should the water concentration fall below 1 gpm per sq ft of tower area, it will be necessary to employ the next higher tower size. This is because in extremely light water concentrations the water will not be distributed uniformly and the performance as predicted by the accompanying curves will not be achieved. Should the water concentration exceed 3 gpm per sq ft of tower area, it will then be necessary to choose the next size lower tower. Higher concentrations tend to blanket the

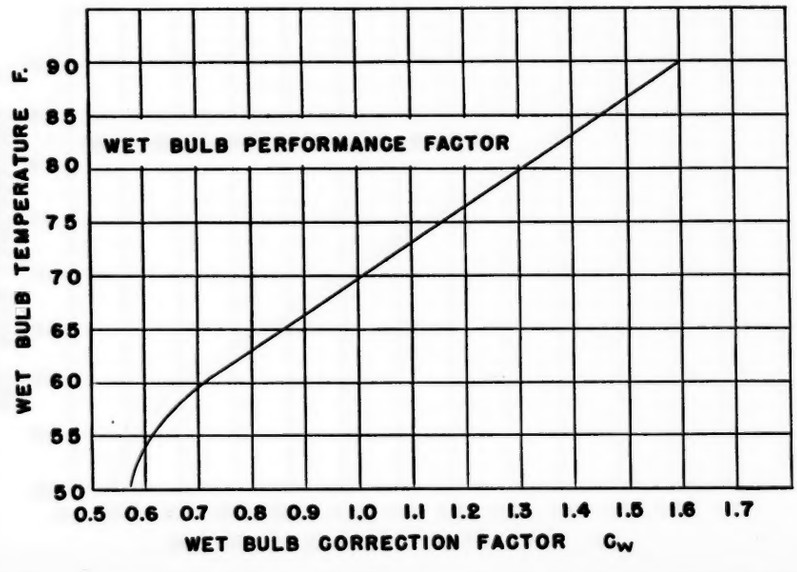


Fig. 3. Wet bulb performance factor.

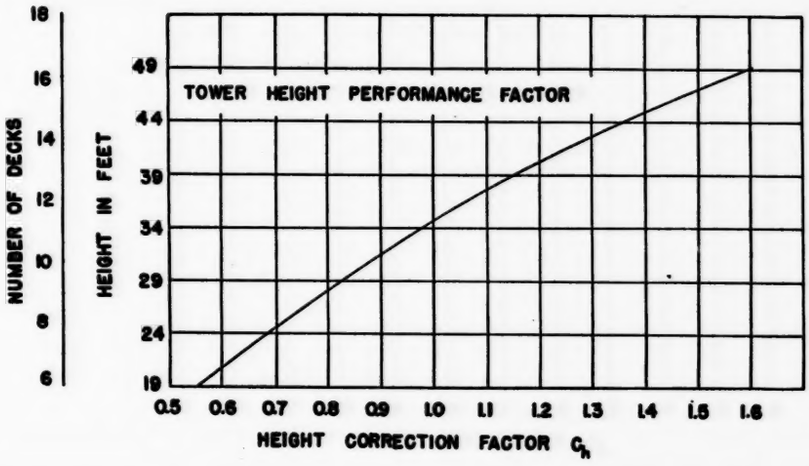


Fig. 4. Tower height performance factor.

tower and will not allow sufficient air to pass through to accomplish the cooling shown on the curves.

(3) *Wet Bulb Temperature.* Theoretically, a cooling tower cannot cool water to a temperature lower than the prevailing wet bulb temperature. With this limit, one is more interested in the economic approach of the cold water temperature to the wet bulb temperature. Air has a greater capacity for absorbing heat at the higher wet bulb temperatures. At the lower wet bulb, air in passing through the tower must have a greater temperature rise to accomplish the same cooling. Therefore, to obtain the same approach at the lower wet bulb temperatures, it is necessary to reduce the concentration.

To calculate the size of an atmospheric type cooling tower with effective width of 12 ft, the following general formula may be used:

$$L = \frac{Q \times W}{C \times 12 \times C_w \times C_h}$$

when:

L — Length of tower in feet.
Q — Quantity of water in gallons per minute.
W — Wind correction factor.
C — Concentration of water per sq foot of cooling tower area.
C_w — Wet bulb correction factor.
C_h — Tower height correction factor.
T_1 — Inlet temperature.
T_2 — Outlet temperature.
$(T_1 - T_2)$ — Temperature range.
T_{wb} — Wet bulb temperature.
$(T_2 - T_{wb})$ — Approach to wet bulb temperature.

PROBLEM 1: Determine the length of a tower 35 ft high to cool 1500 gpm from 90F to 75F with a 70F W.B. and a 3 mph wind. From these conditions we know that the approach $(T_2 - T_w) = 5F$ and the range $(T_1 - T_2) = 15F$. From Fig. 2, it is found that these two values require a concentration of $C = 1.17$. The correction factors, C_h, C_w and W will be equal to one, as shown on their respective correction factor curves, because Fig. 2 was based upon values equal to those given in the problem. Substituting the corresponding values on the above formula we find that a tower 107 ft long is required.

PROBLEM 2: Let us take the same conditions that exist in Problem 1 and see what the effect would be when the wind velocity is increased to 5 mph. From

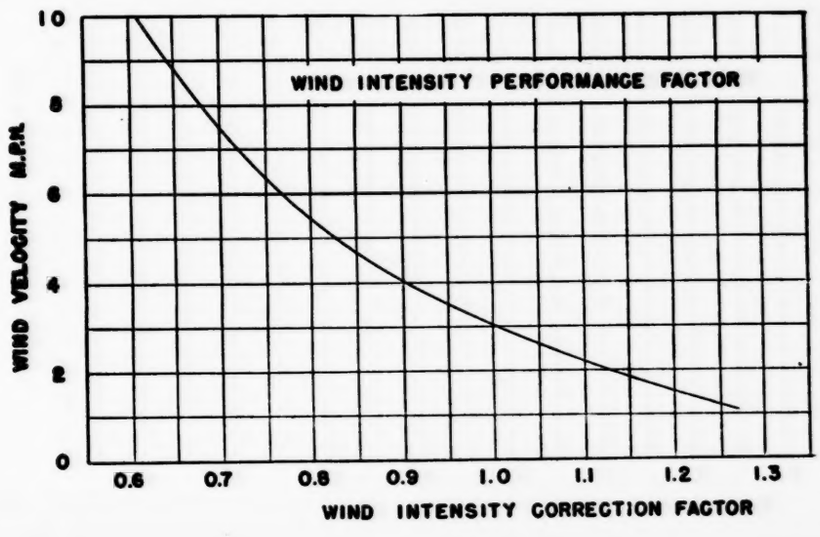


Fig. 5. Wind intensity performance factor.

Fig. 5, when the wind velocity is 5 mph, W = 0.83. Substituting this value for *W* in the formula, we find that a tower 88.5 ft long is required.

PROBLEM 3: Again assume that the same conditions as in Problem 1, except that due to space considerations it is necessary to have a tower 51 ft high. From Fig. 4, when tower height is 51 ft, $C_h = 1.6$. Substituting the new C_h in the formula we find that a tower 65.5 ft long is required.

PROBLEM 4: Assume the conditions of this problem are identical to those of Problem 1, except that water has to be cooled from 85F to 70F and the wet bulb temperature is 65F. From Fig. 3, when the wet bulb temperature is 65F, $C_w = 0.86$. Substituting this factor in the general formula we find tower length to be 124 ft.

PROBLEM 5: Once a tower is installed the problem often arises of determining what cold water temperature one can expect under conditions differing from those for which the tower was designed. For example, let us take the tower calculated for Problem 1. This tower is 35 ft high, 12 ft wide, and 107 ft long. What cold water temperature T_2 can be expected when the wet bulb temperature is 60F, wind velocity 4 mph, ccoling range 15F, and water circulation 2,000 gpm?

The wet bulb temperature correction factor C_w from Fig. 3 is 0.71. The wind velocity correction factor (W) from Fig. 5 is 0.905. By substituting those values in the general formula and solving for water concentration, we find:

$$C = \frac{2{,}000 \times .905}{107 \times 12 \times 0.71 \times 1} = 1.98 \text{ gal per sq ft.}$$

Turning to Fig. 2, we find that when the cooling range is 15F and the water concentration is 1.9, the approach to the wet bulb temperature is 9F. This means that the cold water temperature will be

$$60\text{F} + 9\text{F} = 69\text{F}.$$

The general information given is presented solely for the purpose of showing the more important factors that affect the design of atmospheric type cooling towers. Space does not allow consideration of the effect of excessive or insufficient concentration, higher relative humidity, change in wind direction, and other finer points which must be analyzed for proper cooling tower selection. The cooling tower sizes calculated in the examples are only approximate. This survey, while presenting the essentials, has purposely been kept as non-technical as possible in order that it may serve as a guide to evaluating cooling tower performance.

Radical Fuel Nozzle Development

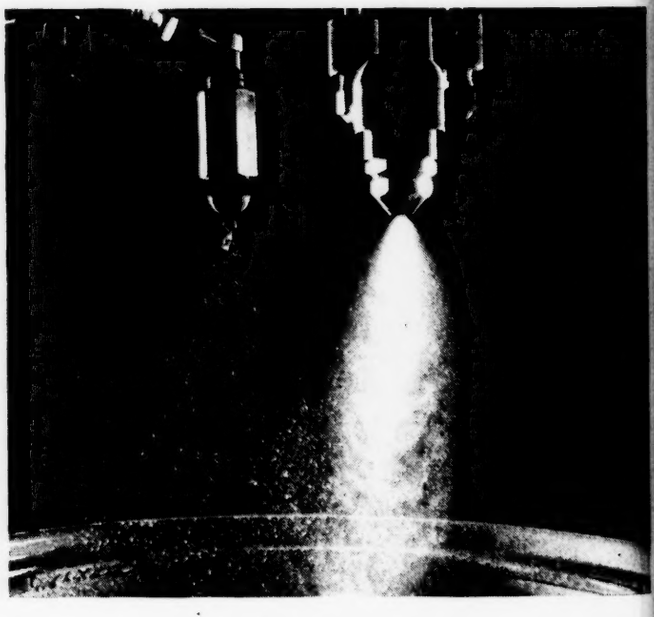
A fuel nozzle recently perfected by Keith V. Smith of the Westinghouse Research Laboratories has the unique advantage of giving superior atomization at all fuel rates—even those approaching zero. It performs effectively below the drizzle point of the conventional mechanical atomizing nozzle, and unlike the ordinary nozzle which sprays larger droplets in a hollow cone pattern, it sprays a solid cone of heavy fog.

Developed in connection with gas turbine combustion studies, it will be used on the 2000-horsepower gas turbine generator set now under experimental development at the Westinghouse South Philadelphia Works, which has been designed to run on No. 6 (Bunker "C") fuel oil. The experimental gas turbine is intended for locomotive, marine and utility applications.

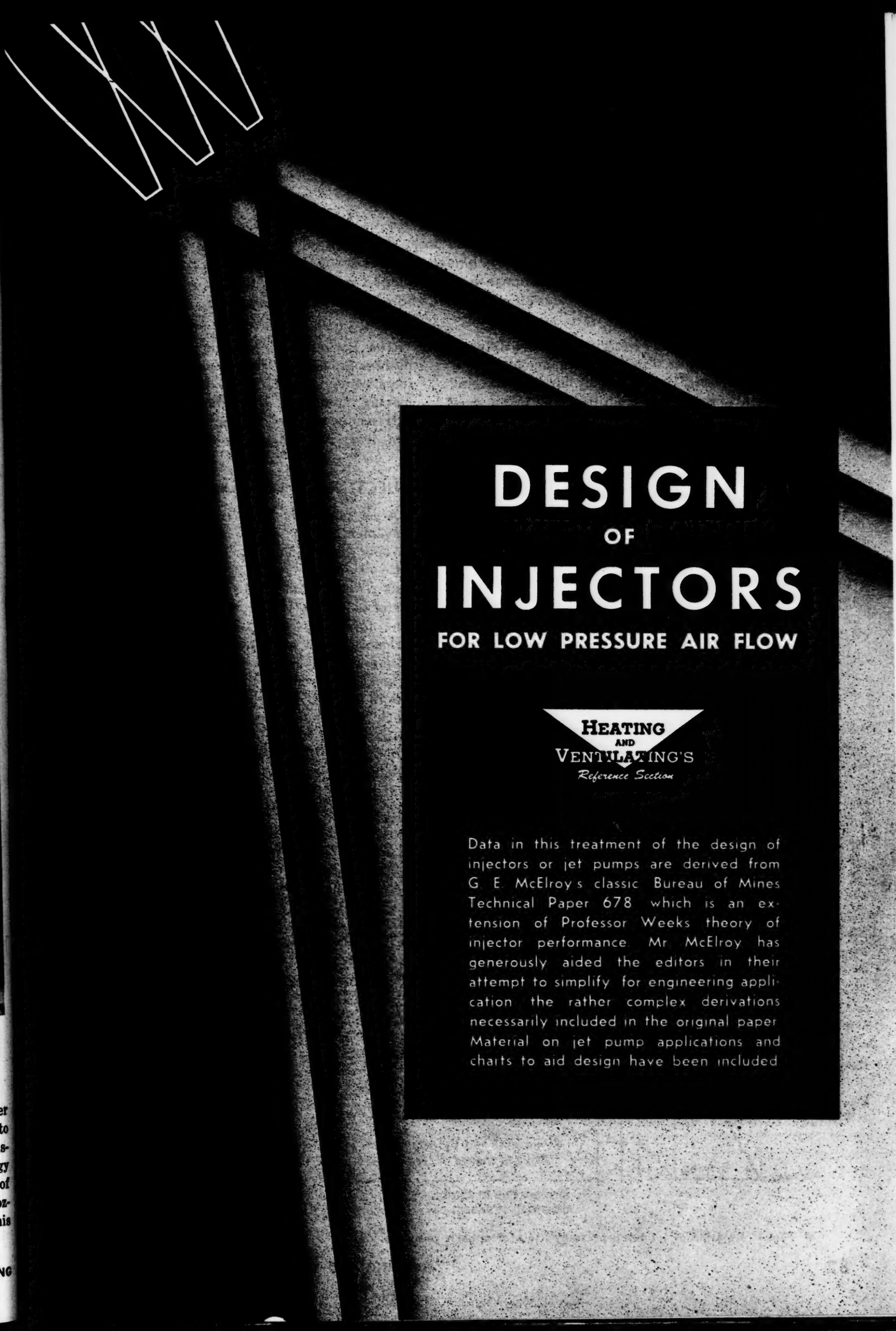
Key to the new nozzle's atomizing power is the very small flow of relatively low-pressure compressed air—less than 1 cfm—that enters through six tiny airducts cut at an angle in the nozzle. As these separate streams of air enters the atomizer, their whirling motion sets up a miniature cyclone. This air blast collides with the incoming fuel supply, ripping the heavy oil into droplets so fine that they approach the size of fog particles.

Two obstacles to the use of conventional nozzles in gas turbines are overcome in this manner: (1) unburned carbon particles are prevented from caking on the combustion chamber walls by the even distribution of finely divided particles throughout the chamber and (2) superfine atomization over a wide range of operation largely prevents partial combustion due to the large size of droplets.

With the new nozzle very fine spray can be attained at exceedingly low oil pressures. However, 10 lb per sq in. has been assumed as the minimum pressure to prevent slight burner pulsations from influencing discharge. It has been found that 90-96% of the energy of the fuel is converted into heat over a wide range of speeds and loads, whereas with conventional fuel nozzles, combustion efficiency falls off to about half this value under certain conditions.



Smith nozzle, right, and conventional nozzle under fuel pressure of 5 psi.



DESIGN OF INJECTORS FOR LOW PRESSURE AIR FLOW

HEATING AND VENTILATING'S
Reference Section

Data in this treatment of the design of injectors or jet pumps are derived from G. E. McElroy's classic Bureau of Mines Technical Paper 678 which is an extension of Professor Weeks theory of injector performance. Mr. McElroy has generously aided the editors in their attempt to simplify for engineering application the rather complex derivations necessarily included in the original paper. Material on jet pump applications and charts to aid design have been included.

Design of Injectors for Low-Pressure Air Flow

Action of a jet as prime mover in an enclosed air stream is discussed from the point of view of the ventilation engineer. A means of predicting injector performance is developed and a basis for injector design established and augmented by charts.

INJECTOR, ejector, and ejector-injector are different names for the same thing—a jet pump formed by a jet of high velocity fluid (usually air) discharging into, and entraining lower velocity fluid in an open duct or channel. The effect of function of a jet pump—whether it is operated as an injector, ejector or injector-ejector—on its performance is comparable to the effect upon the performance of a fan of its use as a blower, exhauster, or booster. A diagram of the comparative pressures involved is shown in Fig. 1. Because of similarity of basic performance, whatever the function, the term "injector" is used in this article as synonymous with "jet pump."

Applications

Injectors may be used for exhausting corrosive, hot or sticky gases without exposure of a fan rotor or motor to the exhaust stream, or for exhausting explosive gases from enclosed spaces such as tanks, tank cars, or gas holders without contact of the exhausted gas with possible sparks from the fan rotor. They may also be used to ventilate through openings, such as tunnels, without obstructing passage through the openings.

The low efficiency in the use of power of most designs is offset by the simplicity and low cost of the installation. Injectors can be used economically for moving fluids and light materials where power requirements are low or the service required is of a temporary, intermittent or emergency nature.

INJECTOR PERFORMANCE

A jet pump for low-pressure flow can be designed conveniently only as a blower, even though it is to be operated as an exhauster or booster, unless the ratio of total flow to jet flow is known or assumed. This requires that performance be stated in terms of performance in the blowing position, as is usually done for fans. And, as in the case of fans, performance as an exhauster or booster can be deduced from performance as a blower, though not as readily. It is also convenient to rate performance of air injectors in terms of air at some standard density, from which performance at any other density may be calculated.

It has been demonstrated that a jet expands in the open atmosphere at constant static pressure and constant momentum. As the mass of the conical jet increases due to entrainment of the surrounding atmosphere, the velocity diminishes at such a rate that the product of mass per unit time and velocity remains constant. During this free expansion, the total pressure of the jet, which is all in the form of velocity pressure, decreases rapidly.

Fundamentally, the performance of an injector depends on the fact that a high-velocity jet of air loses only a portion of its energy in free expansion when it is directed into an open-end enclosure such as a Venturi or cylindrical duct, and the remaining energy of the jet creates flow through the system to which the injector is attached. Beyond the section at which entrainment is completed, the stream flow of combined jet air and entrainment air is the same as in any flow system.

The whole object of injector

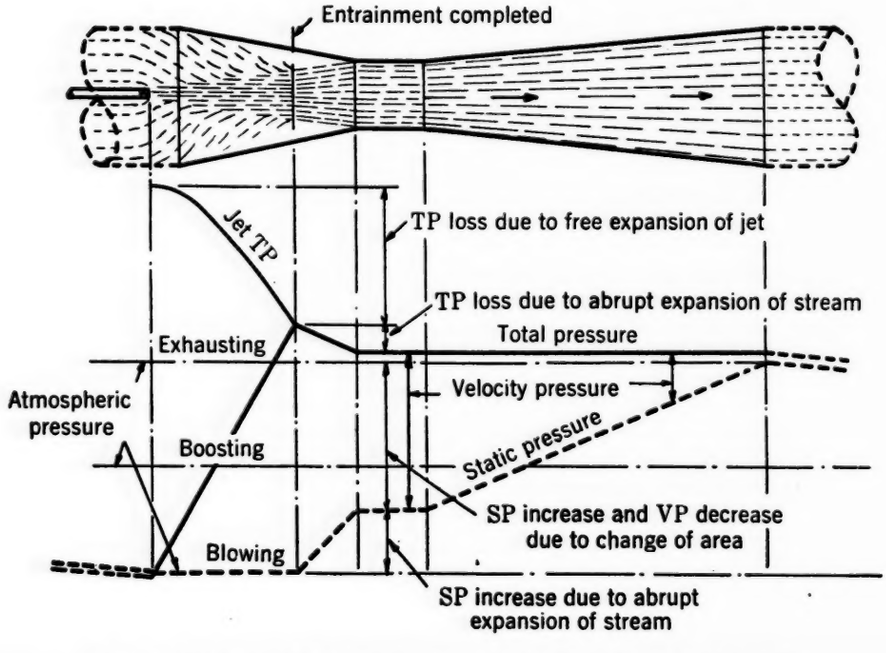


Fig. 1. Relation of jet-pump pressures to atmospheric pressure.

theory therefore is to provide methods for calculating the proportion of jet-velocity energy that remains in the mixed stream after free expansion has taken its toll, and, what is more important, the proportion of static pressure to velocity pressure at discharge as affected by energy losses in stream flow and by mutual conversion of velocity and static pressures.

If the area of the stream at complete entrainment is the same as that of the injector at the section where complete entrainment occurs, and minor friction and shock losses are disregarded, no further energy losses occur. If the area of the stream at complete entrainment is less than that of the injector at that point, as in Fig. 1, abrupt expansion of the combined stream takes place with an energy loss which causes a further decrease in total pressure accompanied by an increase in static pressure. Thus there are two possible causes of energy loss—the original expansion of the jet and, in some cases, the abrupt expansion of the mixed stream. In order to allow the injector to work efficiently against normal static pressures, a diverging outlet is used to convert velocity energy to pressure energy and this, with the converging entrance section, gives the injector its Venturi shape. From this it can be seen that the position of the jet and the form of the channel through which the mixed stream flows are important considerations in injector performance.

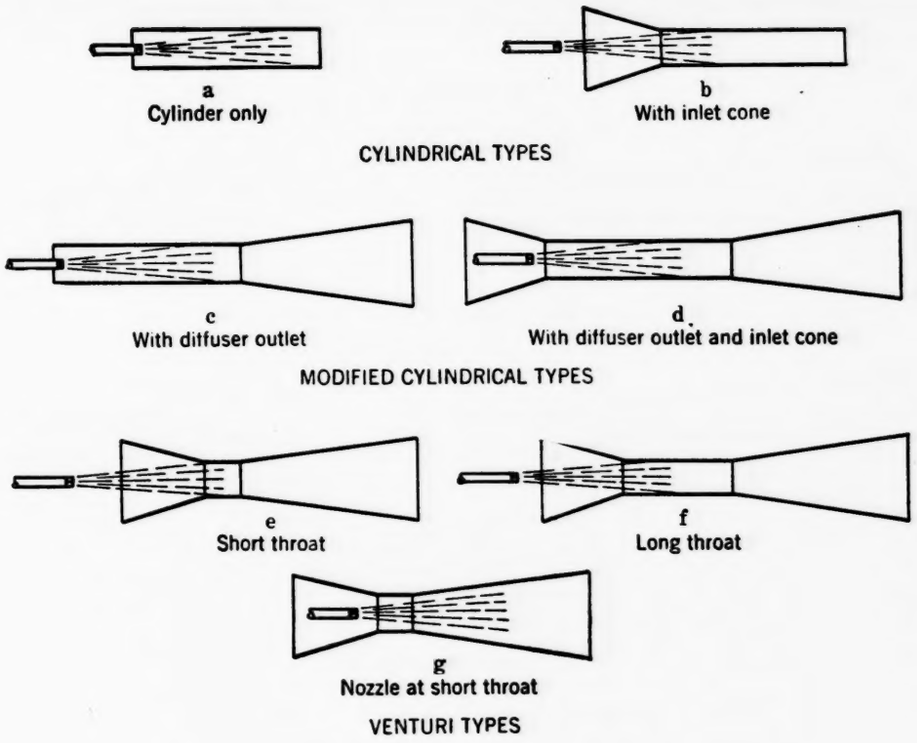


Fig. 2. General types of injectors.

Injector Types

Injectors can be classed in three general groups, shown in Fig. 2, as determined both by the form of the channel and the position of the jet. These groups are: cylindrical types, modified cylindrical types, and Venturi types. The second grouping is required to include cylindrical types to which a diffuser outlet has been added and Venturi types in which the nozzle is placed at the entrance to a long throat. For these two forms, the pressure performance is the same as that of the Venturi type, but the maximium capacity is limited to that of the cylindrical type.

If the nozzle is placed at or close to the throat of a short-throat design, Fig. 2, *g*, the performance cannot be calculated theoretically because the exact section in the diffuser to which abrupt expansion of the stream occurs in not known.

Venturi Injectors

The design of the injector shown in Fig. 2, *e*, with either a longer or shorter throat but with the nozzle set for best performance and operating in the blowing position, is the subject of the discussion in this paper, except where specifically noted otherwise.

The performance of a Venturi injector depends largely on the position of the nozzle. The theory here developed from the original work of Professor W. S. Weeks[1] applies to this design in which the nozzle is so placed ahead of the throat that expansion occurs to throat area or to a larger area in a converging inlet. The length of the throat is of no consequence theoretically.

As the resistance to flow of the system to which the injector is attached decreases, entrainment automatically increases as does also the angle of expansion of the jet. In a cylindrical injector the area at complete entrainment increases until it coincides with that of the cylinder. Up to this point, where the momentum of the stream at the throat equals the momentum of the jet, the total pressure developed by a Venturi is the same as for a cylindrical injector the same size at the throat of a Venturi. Assuming no energy loss by conversion of pressures in the diffuser of the Venturi, the total pressure at discharge is the same as at the throat, but the proportion of velocity pressure to static pressure is different. With a diffuser, equilibrium can be established for static pressure at throat less than static pressure at discharge.

With the nozzle ahead of the throat of a Venturi injector, entrainment can increase to the point where the momentum of the stream equals that of the jet at a section ahead of, and larger than the throat, Fig 3, *e*, at which point the static pressure is the same as that at the nozzle, and the total pressure equals this static pressure plus the velocity pressure of the stream. The stream then contracts to pass the throat and expands again in the diffuser, the overall effect, with a properly positioned nozzle, being the same as gradual

[1]Weeks, W. S., The Air Injector for Auxiliary Ventilation Underground: Engineering and Mining Journal, vol. 138, 1937, pp. 196-199.

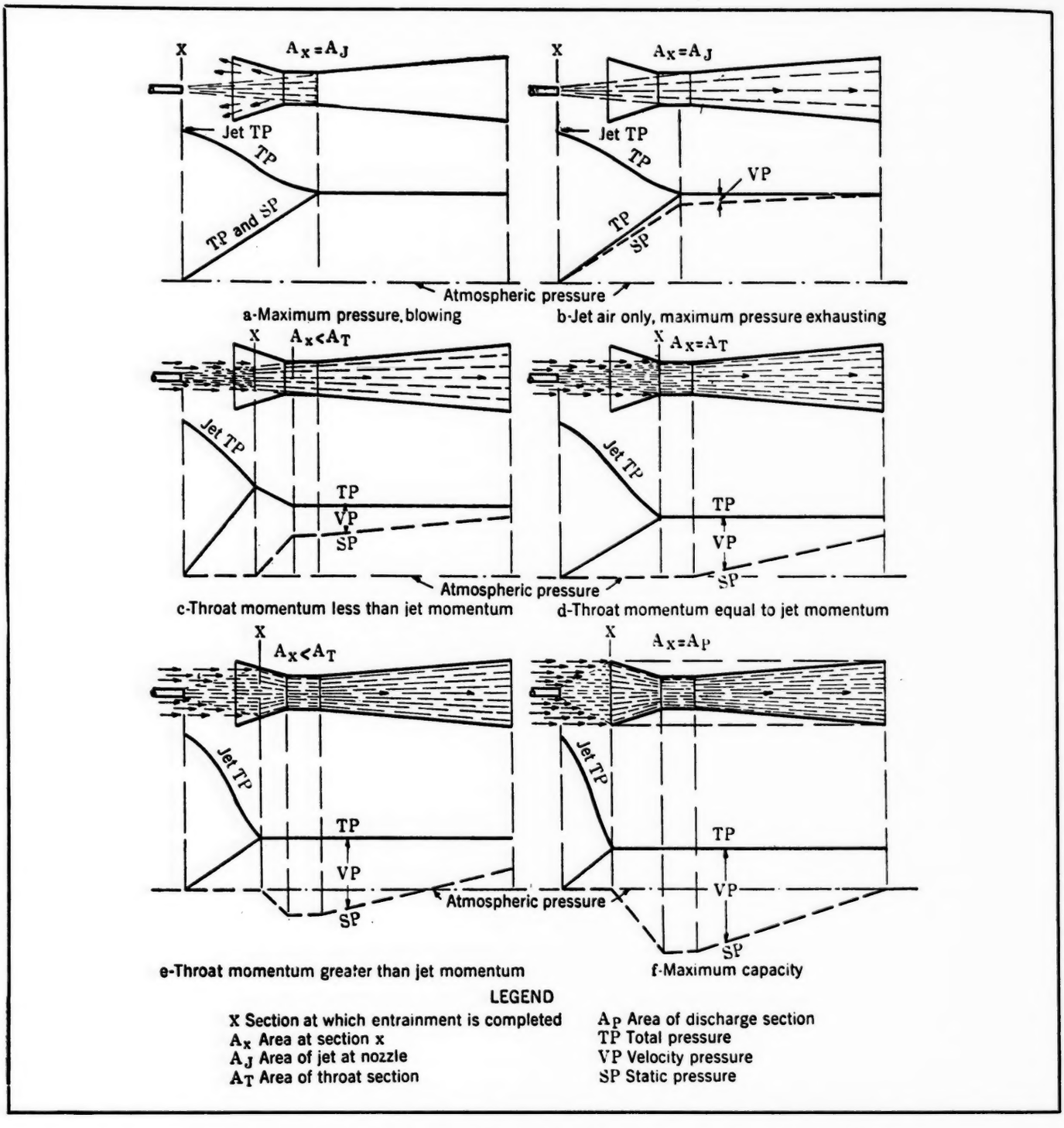


Fig. 3. Diagrams showing typical stages in the performance of a Venturi injector.

expansion in a diffuser direct from the inlet section to discharge. Under these conditions, total pressure remains constant while the static pressure decreases in the throat by the amount that the velocity pressure increases in the converging inlet and then increases in the diffuser by the amount that the velocity pressure therein decreases.

At maximum capacity, Fig. 3, *f*, these two conversions are of equal amount and the static pressure at the throat is less than that at the nozzle and that at the discharge by the amount of static pressure developed from velocity pressure by conversion in the diffuser. Maximum capacity is reached when the momentum of the stream equals that of the jet at a section of area equal to the discharge area of the diffuser, beyond which condition equilibrium cannot be established. Maximum capacity of a Venturi injector is therefore the same as that of a cylindrical injector the same size as the discharge area of the Venturi diffuser.

Pressure Changes

There are two general phases of Venturi injector performance. In one phase the jet is such that the area of the stream at the section of complete entrain-

ment is less than the throat area, resulting in abrupt expansion and a stream momentum less than jet momentum. In the other phase, the area at complete entrainment is equal to or greater than the throat area, resulting in throat momentum equal to or greater than jet momentum.

Graphs of pressure changes for the two general phases of Venturi injector performance are shown in Fig. 3, *c*, for stream momentum less than jet momentum, and in Fig. 3, *e*, for stream momentum equal to jet momentum at a section larger than the throat. Similar graphs for the three limiting conditions of maximum pressure blowing, design capacity (throat momentum equal to jet momentum), and maximum capacity are shown in Fig. 3, *a*, *d*, and *f*, respectively. The special conditions for maximum pressure exhausting are shown in Fig. 3, *b*.

The sketches indicate that entrainment is completed at section *X*, and the stream then expands abruptly or contracts gradually to the throat section. Experiments with air flow in ducts have shown that the increase in static pressure accompanying abrupt expansion in an air stream extends over some distance. Nozzle position in a Venturi injector undoubtedly determines whether or not the maximum pressure increase possible is obtained under the prevailing conditions of design of casing and expansion of jet.

Performance Factors

It is apparent from the pressure relationships charted in Fig. 3 that there are many limiting factors affecting performance. These factors are:

(1) The section at which entrainment is completed and the area of this section;
(2) Momentum of the jet and its area;
(3) Momentum of the mixed stream and its area at various sections of the system;
(4) Whether or not abrupt expansion of the mixed stream occurs, and the pressure changes such expansion brings about;
(5) Velocity pressure at any point in the system;
(6) Conversion of velocity pressure to static pressure in the diffuser outlet;
(7) Efficiency of the system;
(8) Quantity of air entrained under a given set of conditions;
(9) Total and static pressures developed by the injector.

Entrainment Area. The section at which entrainment is complete for a given jet obviously depends upon the position of the nozzle and the angle of expansion of the jet under given conditions. Present data are insufficient to allow calculation of the angle of jet expansion. Therefore the position of the jet, perhaps the most important single feature of the Venturi injector, must be determined by trial for each installation.

This does not mean that the whole structure of injector design is a matter of trial and error. In fact it is merely necessary to design to an assumed section *X* and move the nozzle until performance corresponds to design conditions, bearing in mind that at the section *X* the momentum of the stream equals that of the jet ($M_x = M_j$). It will be found that theoretical and actual performances follow very closely.

The area of the section at which entrainment is completed, A_x, is determined by the cross sectional area of the Venturi at the point *X* except, as in the cases of Fig. 3, *a*, *b*, and *c*, where throat area is greater than A_x.

Air Stream Momentum. The movement of air through the injector depends on the momentum of the jet. Since this involves the weight of a continuous flowing stream, the expression for momentum of an air stream must imply momentum per unit time.

$$M = \frac{WV}{g} = \frac{dQV}{g} = \frac{dAV^2}{g} = \frac{dQ^2}{Ag} \qquad (1)$$

where consistent units require that

M = momentum per second, pounds;
W = weight rate of flow, pounds per second;
V = velocity of flow, feet per second;
d = density, pounds per cubic foot;
Q = volume rate of flow, cubic feet per second;
A = area, square feet;
g = acceleration due to gravity, feet per second per second

Corresponding units for pressure are:

TP = total pressure, pounds per square foot;
SP = static pressure, pounds per square foot;
VP = velocity pressure, pounds per square foot.

A convenient formula for computing the momentum of a low pressure air stream of density *d* lb per cu ft flowing at a rate of *q* cfm is

$$M = \frac{d}{.075} \times \frac{6.48 \times 10^{-7} q^2}{A} \qquad (2)$$

Abrupt expansion. Abrupt expansion of an air stream causes an abrupt loss of momentum and a static pressure rise between a section *X* (Fig. 3, *c*), where the static pressure is that of the jet, and the tube or throat to which expansion occurs. The increase in static pressure over this section, SP_t, in pounds per square foot is (with subscripts *t* for throat and *j* for jet).

$$SP_t = \frac{M_j - M_t}{A_t} \qquad (3)$$

or, in inches of water,

$$sp_t = \frac{M_j - M_t}{5.2A_t} \qquad (4)$$

Velocity Pressure. A convenient formula for computing velocity pressure of an air stream in inches of water, *vp*, for air of density *d* pounds per cubic foot flowing at a rate of *q* cubic feet per minute is

$$vp = \frac{d}{.075} \times \frac{6.23 \times 10^{-8} q^2}{A^2} \qquad (5)$$

Conversion of Pressure. The rise in static pressure due to conversion of velocity to static pressure, $SP_{(conv.)}$, in the diffuser section of a Venturi (neglecting fric-

tion and shock losses) is equal to the difference in velocity pressures between throat and discharge and may be expressed in terms of the velocity pressure at discharge (with subscript p for discharge and other notation as previously given):

$$SP_{(conv.)} = \left[\left(\frac{A_p}{A_t}\right)^2 - 1\right] VP_p,$$

in pounds per square foot (6)

$$\text{or, } sp_{(conv.)} = \left[\left(\frac{A_p}{A_t}\right)^2 - 1\right] vp_p,$$

in inches of water (7)

The same formulas can be used for the net increase in static pressure between section X and the discharge due to the combined effect of conversion of static to velocity pressure in the converging inlet followed by conversion of velocity to static pressure in the diffuser by replacing A_t with A_x.

Efficiency. The efficiency of an injector is the ratio of total energy imparted to the stream divided by kinetic energy of the jet. For pressures in inches of water and quantities in cubic feet per minute,

Stream energy $= 5.2\ tp \times q$ foot-pounds per minute.

Jet energy $= 5.2\ vp_j \times q_j$ foot-pounds per minute.

$$E = \frac{tp \times q}{vp_j \times q_j} \quad (8)$$

For high-pressure jets, it is more convenient to use weight flow, W_j, in pounds per second and velocities, V_j, in feet per second, in which case,

$$\text{Jet energy} = 60 \times \frac{W_j V_j^2}{2g} \text{ foot pounds per minute,}$$

and

$$E = \frac{5.2\ tp \times q}{60 W_j V_j^2 2g} \quad (9)$$

Quantity of Flow. From design rating (where $A_x = A_t$) to maximum capacity (where $A_x = A_p$), if the area of the section, X in Fig. 3, *c*, at which the momentum of the stream equals that of the jet ($M_x = M_j$), is known or assumed, the rate of flow can be calculated and from it the corresponding pressures. Writing Equation (2) to solve for M_x, substituting M_j, and transposing,

$$q = \sqrt{\frac{A_x M_j}{6.48 \times 10^{-7}} \times \frac{.075}{d}} \quad (10)$$

Design rating is that rating at which maximum quantity is obtained at maximum efficiency for a particular resistance-to-flow condition. Each design will, however, operate at higher efficiency against some higher resistance condition for which the flow will be less than desired. It can be shown that this maximum efficiency, as distinguished from design efficiency, occurs in the range of operation represented by Fig. 3, *c*, where the momentum of the stream is less than that of the jet and at the point where the static pressure rise between jet and throat equals the velocity pressure at the throat, that is, $SP_t = VP_t$.

The rate of flow at maximum efficiency can be calculated from Equations (2), (4), and (5) and is

$$q = \sqrt{\frac{A_t M_j}{9.72 \times 10^{-7}} \times \frac{.075}{d}} \quad (11)$$

Total Pressure. From rating to maximum capacity, the total pressure developed by an injector is equal to the velocity pressure at a section, X in Fig. 3, *e*, at which the momentum of the stream equals that of the jet ($M_x = M_j$), that is,

$$TP_p = VP_x \text{ or } tp_p = vp_x$$

From Equations (2) and (4), it can be shown that in this range,

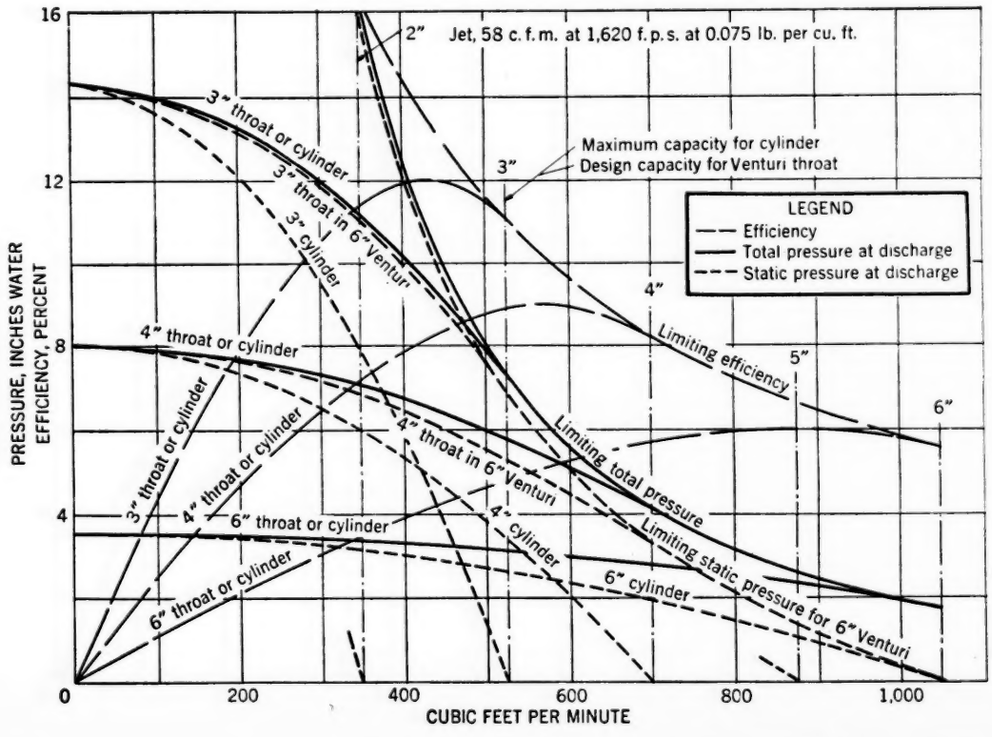


Fig. 4. Characteristic performances of same jet in different injectors.

Table 1—Theoretical Performance of a 6-in.-diameter (0.1963-sq ft) Injector

having a 3-in.-diameter (0.0491-sq ft) throat, when operated by a jet discharging 4.33 lb of air per minute at a velocity of 1,620 fps into air weighing 0.075 lb per cu ft.

Operating conditions		q, cfm	M_t, lb	3.64—M_t, lb	sp_t, in. wg	vp_t, in. wg	tp_p, in. wg	vp_p, in. wg	sp_p, in. wg	Stream energy, ft-lb per min.	E × 100, percent
$M_t < M_j$		0	0	3.636	14.24	0	14.24	0	14.24	0	0
		200	0.578	3.058	11.97	1.03	13.00	0.064	12.94	13,520	7.64
	$sp_t = vp_t$	429	2.429	1.207	4.72	4.72	9.44	.297	9.14	21,050	11.89
							(vp_x)				
$M_t = M_j$	$A_t = A_x$	525	(3.636)	(0)	(0)	(7.12)	7.12	.445	6.67	19,430	10.98
		600	.....	.....	(−3.85)	(9.30)	5.45	.582	4.87	17,000	9.61
		800	.....	.....	(−13.38)	(16.44)	3.063	1.034	2.03	12,750	7.20
	$A_p = A_x$	1,050	.....	.....	(−26.72)	(28.50)	1.781	1.781	0	9,720	5.49

NOTE: Values in parentheses not required for plotting performance but added to show pressures at throat.

$$tp_p = vp_x = \frac{.075}{d} \times \frac{1.483 \times 10^5 M_j^2}{q^2} \qquad (12)$$

From shutoff to rating, where momentum at the throat is less than jet momentum, the total pressure at discharge is equal to the total pressure at the throat, and

$$tp_p = tp_t = sp_t + vp_t$$

which may be calculated from Equations (2), (4) and (5).

Performance Curves

The theoretical performance of an injector operating as a blower can be calculated by means of the formulas given, and the results can be plotted in the form of characteristic pressure — quantity — efficiency curves similar to those used to express the performance at constant speed of a fan operating as a blower.

Two groups of calculations are required to cover the full range of operation from shutoff to wide open, that is from maximum pressure to maximum capacity. The first group covers operation when throat momentum is less than jet momentum ($M_t < M_j$). The second group covers operation when throat momentum is equal to or greater than jet momentum ($M_x = M_j$ and A_x lies between A_t and A_p).

Example. Calculate the theoretical performance of a 6-in. diameter (.1963 sq ft) injector with a 3-in. (.0491 sq ft) throat, operated by a jet discharging 4.33 lb of air per minute at a velocity of 1,620 fps into air weighing .075 lb per cu ft.

Solution. From the data given, the following general values can be derived:

from Equation (1)

$$M_j = 3.636 \text{ lb}$$

from Equation (9)

Stream energy $= 5.2 \times tp_p \times q$

Jet energy $= 177{,}000$ ft-lb per min.

$$E = \frac{5.2 \times tp_p \times q}{177{,}000}$$

from Equation (2)

$$M_t = 1.32 \times 10^{-5} q^2$$
$$M_p = 3.30 \times 10^{-6} q^2$$

from Equation (5)

$$vp_t = 2.584 \times 10^{-5} q^2$$
$$vp_p = 1.616 \times 10^{-6} q^2$$

For the range $M_t < M_j$ where

$$tp_p = tp_t = sp_t + vp_t,$$

assumed quantities are used to calculate vp_t and M_t. The latter is then used to calculate sp_t,

$$sp_t = \frac{3.636 - M_t}{.2553} \qquad \text{[Equation (4)]}$$

At maximum efficiency,

$$sp_t = vp_t \text{ and}$$

$$q = \sqrt{\frac{.0491 \times 3.636}{9.72 \times 10^{-7}}} = 429 \text{ cfm.} \quad \text{[Equation (11)]}$$

For the range $M_x = M_j$, where

$$tp_p = vp_x$$

assumed quantities are used in

$$vp_x = \frac{1.961 \times 10^6}{q^2} \qquad \text{[Equation (12)]}$$

For known areas, A_x,

$$q = \sqrt{\frac{A_x \times 3.636}{6.48 \times 10^{-7}}} = 2369\sqrt{A_x} \quad \text{[Equation (10)]}$$

Thus, there are three points of operation at which quantities of flow can be calculated. In the first range is the point at maximum efficiency where q = 429 cfm. In the second range are the extremes from rating, where $A_x = A_t$, to maximum capacity where $A_x = A_p$. These areas are known, and the quantities 525 cfm at rating and 1,050 at maximum capacity can be calculated from the above equation based on Equation (10). Substituting other assumed quantities within the indicated ranges, values can be tabulated as in Table 1, and the resulting pressures and efficiencies plotted. The indicated curves in Fig. 4 are plotted from these figures.

Limiting Pressure and Efficiency Curves. If, for the same jet and injector discharge areas used in the previous example, a series of performance curves for various throat sizes is calculated and plotted, as has been done in Fig. 4, it is seen that any jet has limiting pressure and efficiency characteristics that are not dependent on the discharge area. This area does, however, determine the static pressure at discharge and maximum capacity.

Fig. 4 also shows that the pressure and efficiency performance curves branch off from the curves of limiting values at points corresponding to rating. These points occur at slightly lower than peak efficiencies, but a throat area based on the flow at peak efficiency would produce another curve with even greater efficiency at a still lower rate of flow. For instance a 3-in. throat gives a maximum efficiency of about 12% at 429 cfm, but is rated for 525 cfm at 11%. This is the maximum efficiency possible for a flow of 525 cfm with the jet used, but an efficiency of 13.5% at rating could be obtained for a flow of 429 cfm with a 2.4-in. throat. This rather anomalous condition is due to the fact that the rate of energy loss is less for abrupt expansion of an air stream than for the free expansion of a jet under the same average velocity conditions.

INJECTOR DESIGN

The design condition of maximum quantity at maximum efficiency requires that the momentum in the throat of a Venturi-shaped injector should equal the momentum of the jet. This condition requires that there be no abrupt expansion of the stream and therefore no change in static pressure between jet and throat. Thus the static pressure at discharge must be developed entirely by conversion of throat velocity pressure in the expanding discharge section.

The required velocity pressure at the throat for the condition of maximum quantity at maximum efficiency must equal the sum of the velocity pressure and static pressure at discharge. This condition is pictured in Fig. 3, *d* and is described by equations,

$$M_j = M_t$$
$$VP_t = SP_p + VP_p = TP_p$$

With velocity pressure at the throat and quantity of flow known, the required area of the throat and throat momentum can be calculated. The jet velocity can be determined from pressure on various types of nozzles. With jet velocity known, the rate of flow required to satisfy the equation $M_j = M_t$ can be calculated. The size of nozzle can then be determined according to the type used.

The momentum required for a specific performance may be obtained by various combinations of jet rate of flow and jet velocity. Just how it is obtained has a determining effect on the resulting efficiency and ratio of entrainment. Entrainment is an inefficient process. Minimum entrainment ratios lead to high efficiency but require large jets. Larger entrainment ratios are less efficient but allow smaller, more compact designs. These and other factors limit injector design, but where there is a wide choice among factors, a design procedure more selective than simple theory will avoid tedious trial and error calculations.

Design Ratios

In the design of Venturi injectors, it is convenient to set up a series of dimensionless ratios which can be used in calculating the relationships existing in any range of design conditions. Since results are expressed in dimensionless ratios, any convenient system of units may be used. In the following analysis, the foot-pound-second system is followed, in which the notation previously used is continued. The dimensionless ratios are:

$$R_q = \frac{Q}{Q_j} = \frac{\text{total volume rate of flow}}{\text{jet volume rate of flow}}$$

$$R_a = \frac{A_t}{A_j} = \frac{\text{area of throat}}{\text{area of jet}}$$

$$R_x = \frac{A_p}{A_x} = \frac{\text{area of discharge}}{\text{area at complete entrainment}}$$

$$R_t = \frac{A_p}{A_t} = \frac{\text{area of discharge}}{\text{area of throat}}$$

$$R_p = \frac{A_p}{A_j} = \frac{\text{area of discharge}}{\text{area of jet}} = R_a R_t$$

$$R_d = \frac{d_j}{d} = \frac{\text{density of jet}}{\text{density of stream}}$$

$$E = \frac{\text{useful energy applied to flow}}{\text{jet energy}} = \frac{\text{efficiency percent}}{100}$$

The analyses are made for low pressure flow, with $Q = Q_x = Q_t = Q_j$ and $d_x = d_t = d_j$. That is, changes in density and volume rate of flow after complete entrainment are ignored.

Shutoff to Rating

These dimensionless relationships in the operating range from shut off to rating can be shown to be as follows:

$$TP_p = TP_t = \frac{2M_j - M_t}{2A_t} \qquad (13)$$

$$E = \frac{2R_q}{R_a} - \frac{R_q^3}{R_a R_a^2} \qquad (14)$$

At Shutoff. For zero quantity, $M_t = 0$ and, from (13), the total pressure is a maximum at

$$TP_{p-max} = \frac{2M_j}{2A_t} = \frac{2(VP_j)}{R_a} \qquad (15)$$

Design Pressure. At the design condition or rating where

$$M_t = M_j$$

$$TP_{design} = \frac{M_j}{2A_t} = \frac{VP_j}{R_a} \qquad (16)$$

It can be seen from Equations (15) and (16) that TP_p at rating is equal to 1/2 TP_p at shutoff for a given injector.

Maximum Efficiency. Considering R_d and R_a constant, the conditions for maximum efficiency for a given ratio of throat size to jet size can be determined as

$$R_q = \sqrt{\frac{R_a R_d}{1.5}} \text{ and } R_a = \frac{1.5}{R_d} R_q^2 \qquad (17)$$

Substituting these values in (13),

$$E_{max} = .889 \frac{R_d}{R_q} \text{ or } E_{max} = 1.089 \sqrt{\frac{R_d}{R_a}} \qquad (18)$$

Design Efficiency. Considering R_q and R_d constant, the conditions for design efficiency (or maximum efficiency at a given ratio of total flow to jet flow) can be determined as

$$R_a = \frac{R_q^2}{R_d} \text{ and } R_q = \sqrt{R_a R_d} = \sqrt{\frac{R_d R_p}{R_t}} \qquad (19)$$

Substituting these values in (13),

$$E_{design} = \frac{R_d}{R_q} \text{ or } E_{design} = \sqrt{\frac{R_d}{R_a}} \qquad (20)$$

Resistance Ratios. If resistance to flow is expressed in terms of velocity pressures, the performance of an injector from shutoff to rating can be expressed in nondimensional ratios of pressures, areas, and rates of flow, as

$$\frac{SP_t}{VP_t} = \frac{2R_a R_d}{R_q^2} - 2 \qquad (21)$$

$$\frac{TP_t}{VP_t} = \frac{2R_a R_d}{R_q^2} - 1 \qquad (22)$$

$$\frac{TP_t}{SP_t} = \frac{TP_p}{R_t^2 VP_p} \qquad (23)$$

At maximum efficiency, where

$$R_q = \sqrt{\frac{R_a R_d}{1.5}}, \frac{TP_t}{VP_t} = 2; \text{ and } VP_t = SP_t.$$

At rating, where

$$R_q = \sqrt{R_a R_d}, \frac{TP_t}{VP_t} = 1; \text{ and } SP_t = 0.$$

To secure operation at rating, it is merely necessary to make

$$R_t = \sqrt{\frac{TP_p}{VP_p}} = \sqrt{\frac{SP_p}{VP_p}} + 1 \qquad (24)$$

Rating to Capacity

Performance of a Venturi injector from rating to wide open lies along the curves of limiting pressure and limiting efficiency. The total pressure developed is the velocity pressure at a section *X*, where entrainment is completed, intermediate in area between the throat area and the area of discharge. As mentioned previously, the stream momentum at this section is equal to the jet momentum. In this range of operation, where $M_x = M_j$,

$$TP_p = VP_x = \frac{M_j}{2A_x} \qquad (25)$$

$$E = \frac{A_j}{A_x}(R_q) = \frac{R_d}{R_q} \qquad (26)$$

$$R_q = \sqrt{\frac{R_d R_p}{R_x}} \qquad (27)$$

and

$$\frac{TP_t}{VP_t} = \left(\frac{R_x}{R_t}\right)^2 \qquad (28)$$

Design conditions. It can be seen that the limiting efficiency has a maximum value for any rate-of-flow ratio and fixed jet when A_x is a minimum, that is, when $A_x = A_t$. Then $R_x = R_t$ and

$$TP_{p-design} = \frac{M_j}{2A_t} = \frac{VP_j}{R_a} \qquad (29)$$

$$R_q = \sqrt{R_a R_d} = \sqrt{\frac{R_d R_p}{R_t}} \text{ or } R_a = \frac{R_q^2}{R_d} \qquad (30)$$

$$E_{design} = \frac{R_d}{R_q} = \sqrt{\frac{R_d}{R_a}} \qquad (31)$$

$$\frac{TP_t}{VP_t} = 1, \text{ so } TP_t = VP_t \text{ and } SP_t = 0$$

and operation at rating is secured by making

$$R_t = \sqrt{\frac{TP_p}{VP_p}} = \sqrt{\frac{SP_p}{VP_p}} + 1 \qquad (32)$$

Comparing Equations (16), (19), (20) and (24) with (29) to (32) it is apparent that the two phases of the performance curve meet in a common point at which the efficiency for a given rate of flow is a maximum for a fixed throat area and jet. The efficiency at this point of operation is termed *design* efficiency to distinguish it from the true *maximum* efficiency obtained at a lower rate of flow than the desired rate.

If the rate-of-flow ratio at maximum efficiency is designated R_{qm} and that at design efficiency R_{qd}, then for the same R_a and R_d,

$$\frac{R_{qm}}{R_{qd}} = .8163 \qquad (33)$$

Maximum capacity. At maximum capacity, $A_p = A_x$, the momentum of the stream at discharge is equal to the momentum of the jet, and

$$R_{q-max} = \sqrt{R_d R_p} = \sqrt{R_d R_t R_a} \qquad (34)$$

Efficiency at maximum capacity is

$$E_{q-max} = \sqrt{\frac{R_d}{R_a R_t}} \qquad (35)$$

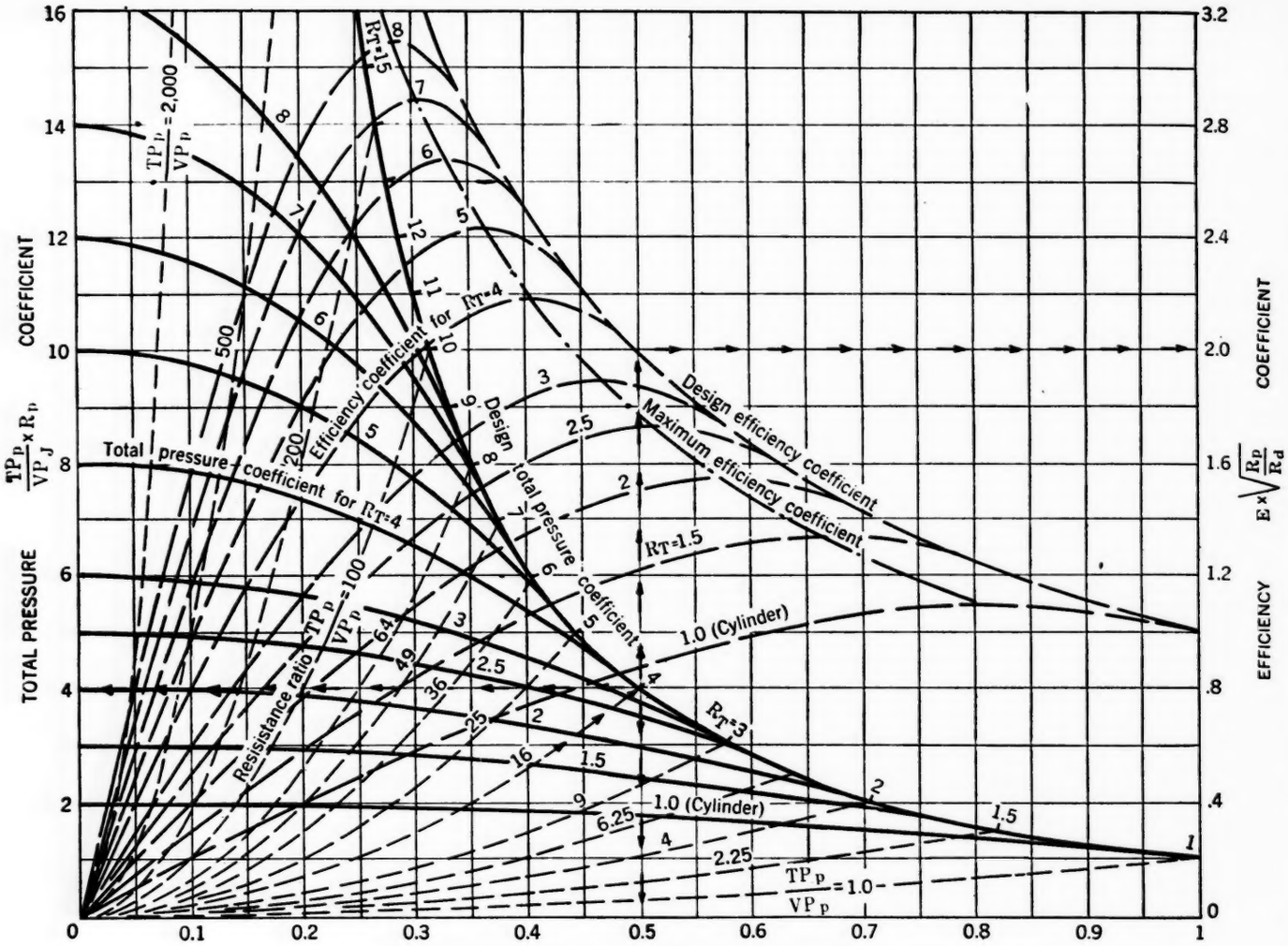


Fig. 5. Characteristic ratio chart of theoretical performance of low-pressure air injectors.

Characteristic Ratio Chart

By the use of particular combinations of the ratios described, set up in the form of pressure, efficiency, and quantity coefficients, generalized curves of injector performance can be plotted. These characteristic ratio curves assume the exact form of performance curves for a particular design and jet when the parameters,

$$\text{total pressure coefficient} = \frac{TP_p}{VP_j} \times R_p$$

$$\text{and efficiency coefficient} = E \times \sqrt{\frac{R_p}{R_d}}$$

are plotted against a

$$\text{quantity ratio coefficient} = R_q \times \frac{1}{\sqrt{R_d R_p}}$$

as in Fig. 5. This chart shows characteristic ratio curves for a range of discharge-area-to-throat-area ratios (R_t) of from 1 to 8.

The coefficients used to plot the chart shown in Fig. 5 were evaluated for assumed pressure and area ratios according to the following formulas, where $z = TP_t/VP_t$:

$$\frac{TP_p}{VP_j} \times R_p = \frac{2zR_t}{z+1} \qquad (36)$$

$$R_q \times \frac{1}{\sqrt{R_d R_p}} = \sqrt{\frac{2}{(z+1)R_t}} \qquad (37)$$

$$E \times \sqrt{\frac{R_p}{R_d}} = z\left(\frac{2}{z+1}\right)^{3/2} \sqrt{R_t} \qquad (38)$$

The resulting curves can be used in injector design as shown in the following example.

Example of Blower Design

Problem. An injector is required to blow 3,000 cfm of air through 1,000 ft, of 16-in. diameter straight iron pipe with a jet of a diameter not greater than one-third the throat diameter.

Data. Density differences between jet and stream are considered negligible, that is $R_d = 1$. Straight iron pipe has a resistance pressure equal to about one velocity pressure for 50 diameters. There are 750 diameters and a resistance pressure of 15 velocity pressures in the specified pipe resulting in a TP_p/VP_p value of 15 + 1 or 16. The velocity pressure for 3,000 cfm of .075 lb per cu ft air flowing through 16-in. pipe (area = 1.396 sq ft) at a velocity of 2,150 fpm is .288

in. wg. Total pressure for the specified pipe, then, will be $16 \times .288 = 4.61$ in. wg.

Solution. Having established the TP_p/VP_p value of 16, the chart in Fig. 5 is entered along the curve representing this resistance ratio. This curve crosses the design pressure curve at its junction with the $R_t = 4$ total pressure curve, establishing the throat area as one-fourth the pipe area. Throat diameter is therefore $\sqrt{1/4} \times 16$ or 8 in. The coordinates of the point of intersection of the resistance ratio curve and the design total pressure curve establish the values of 4 and .5, for the total pressure and quantity ratio coefficients respectively. Similarly, a projection of this point to the design efficiency curve, as shown by arrows in Fig. 5, establishes the efficiency coefficient at 2.

With $R_d = 1$, $R_t = 4$, and $R_p = R_t R_a$ by definition, operating conditions will be:

$$R_q = .5\sqrt{R_d R_p} = .5\sqrt{4R_a} = \sqrt{R_a};$$

$$E = 2\sqrt{\frac{R_d}{R_p}} = 2\sqrt{\frac{1}{4R_a}} = \sqrt{\frac{1}{R_a}}$$

and

$$TP_p = \frac{4VP_j}{R_p} = \frac{4VP_j}{4R_a} = \frac{VP_j}{R_a}$$

Since TP_p is shown to be 4.61. in. wg,

$$VP_j = 4.61\, R_a$$

The relations between these four unknowns, R_q, R_a, E, and VP_j, can be developed also from Equations (16), (19), and (20). If one is assumed, the other three can be calculated. It is possible to design for a specific quantity ratio, jet-throat size ratio, jet efficiency, or jet pressure. In this example, as R_a must be greater than $(3)^2$, we assume the diameter of the jet as 2.5 in. and that it is discharged from a straight pipe nozzle of the same size.

$$\text{Then } R_a = \left(\frac{8}{2.5}\right)^2 = 10.24$$

$$R_q = \sqrt{10.24} = 3.2$$

$$E = \sqrt{\frac{1}{10.24}} = .313 \text{ or } 31.3\%$$

and

$$VP_j = 4.61 \times 10.24 = 47.2 \text{ in. wg.}$$

The required jet flow is $\frac{3{,}000}{3.2}$ or 938 cfm. Its velocity through a 2.5-in. opening (area = .0341 sq ft) would be 27,500 fpm, and the velocity pressure, according to Equation (5) and as shown above, would be 47.2 in. wg. The inlet quantity would be 3,000 — 938 or 2,062 cfm, and the static pressure at discharge would be 4.32 in. wg.

Example of Exhauster Design

Problem. An injector is required to exhaust 3,000 cfm of air through 1,000 ft of 16-in. diameter straight iron pipe with a jet of a diameter not greater than one-third the throat diameter.

Data. The required data are the same as for the preceeding problem except that they apply to conditions at the inlet of the injector, where the static pressure will be $15 \times .288 = 4.32$ in. wg.

Solution. The injector design of the preceding problem will not operate at rating against the same duty as an exhauster. By plotting the performance of the blower design and boosting the jet pressure and quantity as required to increase the inlet volume from 2,062 cfm to 3,000 cfm, it can be determined that a jet volume of 1,151 cfm at a pressure of 66 in. wg would be required and that operation would be at 27% efficiency. To secure performance at rating, it is necessary to design an injector that would operate as a blower against the same resistance pressure but with a different discharge velocity pressure due to the addition of jet volume to line volume.

In this example, as R_a must be greater than $(3)^2$ and the TP_p/VP_p is not known, we assume a value of 10 for R_a, which establishes $R_q = \sqrt{R_a} = \sqrt{10} = 3.16$ for rated performance. If the stream quantity at discharge is 3.16 times the jet discharge, then the quantity induced through the inlet is 2.16 times the jet discharge, and a resistance pressure equal to 15 inlet velocity pressures becomes equivalent to $\left(\frac{2.16}{3.16}\right)^2 \times 15$, or 7.0 discharge velocity pressures for a 16-in. diameter discharge, which is also assumed. The equivalent blower rating is thus a TP_p/VP_p ratio of 7.0 + 1.0, or 8.0. Operation at rating against this resistance ratio requires that $R_t = \sqrt{8} = 2.83$. The correct throat diameter is $\sqrt{\frac{1}{2.83}} \times 16$, or 9.51 in. and the corresponding jet diameter $\frac{9.51}{\sqrt{10}}$, or 3.01 in. The total discharge would be $\frac{3.16}{2.16} \times 3{,}000$, or 4,390 cfm, the jet flow 1,390 cfm and the velocity pressure at discharge $\left(\frac{4{,}390}{3{,}000}\right)^2 \times .288$, or .616 in. The total pressure generated is 4.32 + .62, or 4.94 in., the required jet velocity pressure is 4.94 $\times$ R_a, or 4.94 $\times$ 10 = 49.4 in. The resulting efficiency is $\frac{1}{R_q} \times 100$, or 31.6%.

Comparison of the total pressures, quantities and efficiencies of this design with the blower design example shows that the power requirement of the exhauster design is about 56% larger than that of the blower design. By extending the diffuser of the exhauster design to reduce the loss occasioned by the high discharge velocity, power requirements can be lowered somewhat. For the same velocity of 2,150 fpm at inlet and discharge, a discharge-diameter of 19.4 in. would be required and the power requirement would exceed that of the blower design by about 45%.

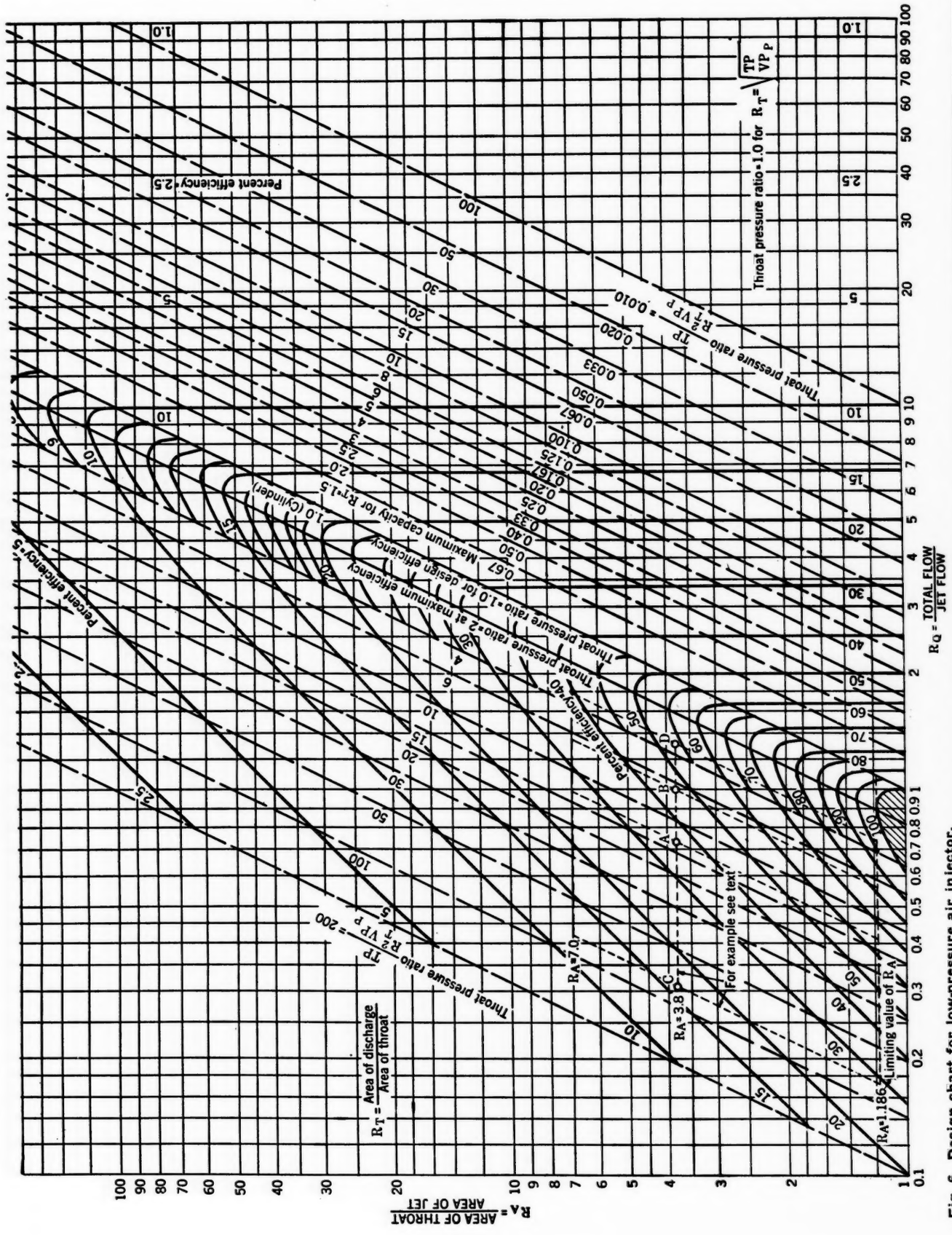

Throat pressure ratio = 1.0 for $R_T = \sqrt{\frac{TP}{VP_P}}$
Throat pressure ratio $\frac{TP}{R_1^2 VP_P} = 0.010$
Throat pressure ratio $\frac{TP}{R_1^2 VP_P} = 200$
Percent efficiency = 2.5
Percent efficiency = 5
Maximum capacity for $R_T = 1.5$
1.0 (Cylinder)
Throat pressure ratio = 1.0 for design efficiency
Throat pressure ratio = 2 at maximum efficiency
Throat efficiency = 40
$R_A = 7.0$
$R_A = 3.8$
$R_A = 1.86$
Limiting value of R_A
For example see text
$R_T = \frac{\text{Area of discharge}}{\text{Area of throat}}$
$R_A = \frac{\text{AREA OF THROAT}}{\text{AREA OF JET}}$
$R_Q = \frac{\text{TOTAL FLOW}}{\text{JET FLOW}}$

Fig. 6. Design chart for low-pressure air injector.

With a negligible velocity pressure at discharge, the excess power requirement would still be 37.5%.

Design Chart for Air Injectors

A chart showing the field of choice directly in terms of the relations of efficiency to resistance, size and rate-of-flow ratios, as Fig. 6, facilitates the design of injectors for specific duties and the construction of performance graphs for specific designs. For design efficiency, R_a and R_q intersect on the resistance line marked "throat pressure ratio = 1". To make actual performances correspond to this design point, it is necessary to make $R_t = \sqrt{TP_p/VP_p}$. Then all points on the performance curves for any specific design can be determined from the intersections of its R_a line with assumed TP_p/VP_p ratios. The chart covers the range from $TP_p/VP_p = 200\ R_t^2$ to $TP_p/VP_p = .01\ R_t^2$.

The chart shows clearly how the performance of an injector is restricted as respects operation as an exhauster or booster. Operation against resistances that would result in performance in the range to the left of the ordinate $R_q = 1.0$ would result in some jet air being blown back out the inlet. Operation as a blower in the same resistance range results in the same blowing back of jet air, but this condition is not only permissible in many cases where the injector is used as a ventilating device rather than an injecting device, but is a requirement in a design for 100% efficiency.

Design for 100% Efficiency. Because of the peculiar relation of maximum efficiency to design efficiency, as shown by Equations (18) and (20), a design for 100% efficiency must be based on the resistance line marked "throat pressure ratio = 2" with $R_t = \sqrt{TP_p/2\ VP_p}$, for which case $R_a = 1.186$ and $R_q = .889$, that is, about 11% of the jet air is blown back out the inlet.

Examples of Use of Design Chart. The example indicated on the chart in dotted lines refers to an analysis of reported test data on the ventilation of a railroad tunnel by a cylindrical ($R_t = 1.0$) injector with sidewall openings of such size as to make $R_a = 3.8$. Test performances fall along the line $R_a = 3.8$. The design was based on obtaining $R_q = 1.0$ against a normal estimated resistance ratio of $TP_p/VP_p = 6.6$ as shown by point *B*. The actual normal resistance ratio proved to be 13.0 (point *A*) and the range of performance, as affected by train movement and natural draft, from 76 (point *C*) to 3.6 (point *D*). If the resistance had been correctly estimated and the same method of design used, performances would have fallen along the $R_a = 7.0$ line with resulting efficiencies about one-fourth lower — 27% as against 36% for normal operation. With the most efficient design of cylindrical injector having $R_a = 1.186$, efficiencies for normal operation could have been increased to 64.5%. With a Venturi design, the efficiency figure for normal operation could be increased somewhat but a 100% design would require an impracticable degree of constriction to a ⅝-diameter throat in this case.

Consideration of the example of *blower design* previously given shows that, with no restriction as to size of jet, a very much more efficient ventilating device could be designed. With R_t set at 4.0 to make $TP_t/VP_t = 1.0$ when $TP_p/VP_p = 16$ and with R_a set at 10.24 based on the selected jet size, performance would be along the $R_a = 10.24$ line, with rated performance indicated by its intersection with the "throat pressure ratio = 1.0" line. This point determines R_q as 3.2 and the efficiency as 31.3% at rating, as in the solution previously given.

With no restrictions as to size of jet, a 100% efficiency design could be used, that is, a design that would operate theoretically at 100% efficiency at rating. In this case $R_a = 1.186$ and R_t would be set at $\sqrt{8}$ to make $TP_t/VP_p = 2.0$ when $TP_p/VP_p = 16$. Performance would be along the $R_a = 1.186$ line, and rated performance would be indicated by the intersection of this line with the "throat pressure ratio = 2.0" line. This point determines R_q as .89 (11% blowback) and the efficiency as 100%. For 16-in. discharge, the throat diameter would be $\frac{1}{\sqrt[4]{8}} \times 16$, or 9.52 in. and the jet diameter $\frac{1}{\sqrt{1.186}} \times 9.52$, or 8.03 in.

For a pipe flow of 3,000 cfm, the jet flow would be $\frac{1}{.89} \times 3{,}000$, or 3,375 cfm and this flow through a 8.03-in. pipe jet would require a velocity pressure of but 5.73 ins. water for .075 lb per cu ft air density.

Consideration of the example of *exhauster design* previously given shows that a more efficient design could be secured with a lower R_a value, that is, with a larger jet. Where "blowback" introduces no hazard, a value of R_q between 1 and 2 could be assumed such that the size of the required jet would not unduly obstruct the inlet flow. Where "blowback" introduces a hazard, the desirable minimum value of R_q is 2.0, which establishes R_a as 4.0 for a throat pressure ratio of 1.0. With a 16-in. discharge, the velocity of discharge, 4,300 fpm, would be excessively high. Assuming a discharge velocity equal to the inlet velocity at 2,150 fpm, the required discharge diameter is $\sqrt{2} \times 16$, or 22.62 in. The resistance pressure of 15 inlet velocity pressures is then equivalent to 15 discharge velocity pressures. The equivalent blower rating is therefore a TP_p/VP_p ratio of 16 and $R_t = \sqrt{16} = 4.0$. The required throat diameter is $\sqrt{1/4} \times 22.62$, or 11.31 in. and the required jet diameter is $11.31/\sqrt{R_a} = 11.31/\sqrt{4}$, or 5.66 in. The total discharge would be $2 \times 3{,}000$, or 6,000 cfm at a velocity pressure of 2.88 in. The total pressure generated is 4.32 + .29, or 4.61 in. The required jet velocity is $4.61 \times R_a = 4.61 \times 4$, or 18.44 in. And the efficiency for rated performance is $\frac{1}{R_q} \times 100$, or 50%.

From theoretical considerations, it is evident that the calculated performance of an injector is the max-

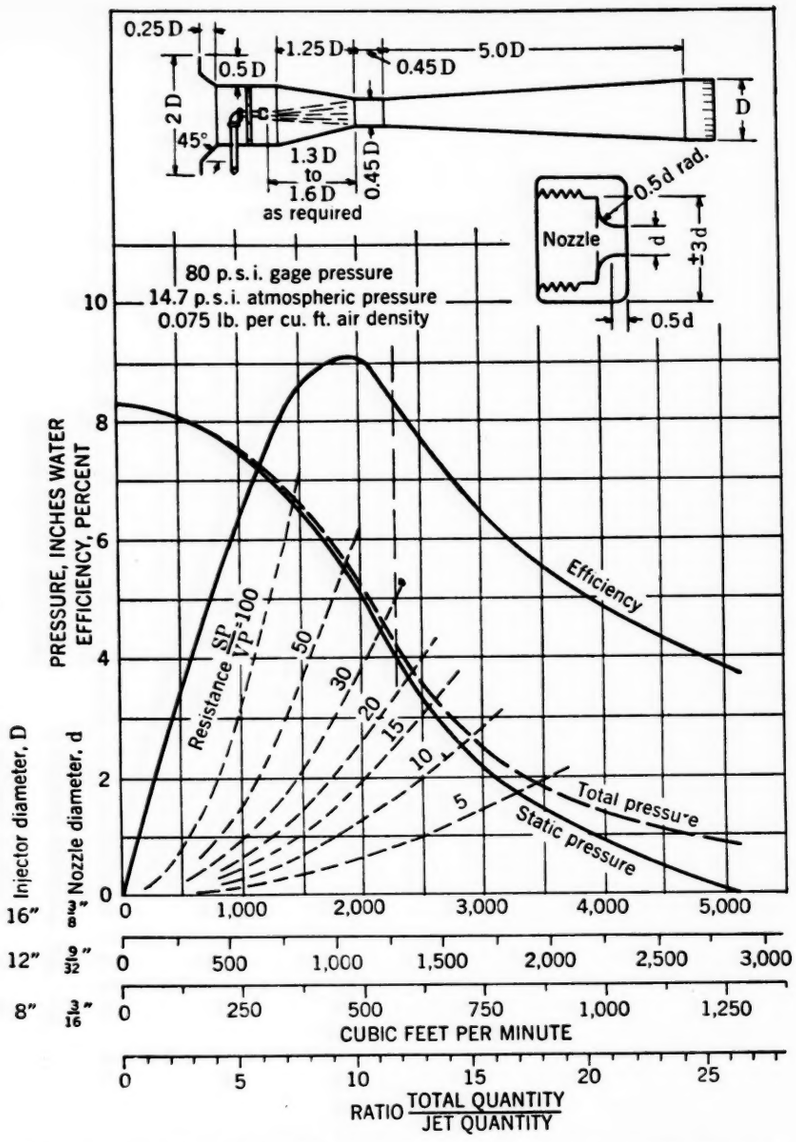


Fig. 7. Performance graphs for homologous series of compressed-air-operated injectors designed for auxiliary mine-ventilation service.

imum possible performance and that it is attained only for a definite relation of the area of the jet, at certain stages in its expansion, to areas of the injector. Experiments show that the angle of expansion varies with the design of the nozzle and the injector and with the operating condition in a manner that, as yet, is not mathematically predictable.

The jet nozzle should be on center and blow along the axis of the injector. For ordinary Venturi designs that include a low-angle diffuser, the best position for rated performance seems to be 3 to 4 throat-diameters ahead of the throat — closer to 3 for compressed-air and sharp-edged orifice nozzles and closer to 4 for straight pipe jets.

Laws of Injector Performance

From the relations that have been given, it is possible to formulate laws for air-injector performance similar to those for centrifugal fan performance. Jet velocity takes the place of fan speed, and throat diameter replaces fan-wheel diameter.

(1) If the jet velocity of an injector operating on a fixed resistance condition is changed (under conditions where the volume rate of jet flow is proportional to jet velocity):

The efficiency will remain constant.

The ratio of total flow to jet flow will remain constant.

The quantity produced will vary directly with the jet velocity.

The pressure developed will vary directly with the square of the jet velocity and directly with the pressure on a jet orifice.

The power required will vary directly with the cube of the jet velocity.

(2) If geometrically similar injectors are equipped with geometrically similar nozzles and operated at the same jet velocity, then for the same resistance ratio of total pressure to velocity pressure at discharge:

The efficiencies, the pressures maintained, and the quantity ratios will be constant.

The quantities produced will vary directly as the square of the ratio of diameters.

The powers required will vary directly as the square of the ratio of diameters.

(3) If geometrically similar injectors are operated by the same nozzle and at the same jet velocity, then for the same operating conditions:

The efficiencies will vary inversely with the ratio of injector diameters.

The quantities produced will vary directly with the ratio of injector diameters.

The pressures maintained will vary inversely with the square of the ratio of injector diameters.

(4) If an injector is operated by geometrically similar nozzles at the same jet velocity, then for the same operating condition:

The efficiencies will vary directly with the ratio of nozzle diameters.

The quantities produced will vary directly with the ratio of nozzle diameters.

The pressures maintained will vary directly with the square of the ratio of nozzle diameters.

Compressed-Air-Operated Venturi Design

An application of the laws of injector performance is shown in Fig. 7 which shows the general design features and performance graphs for an homologous series of compressed-air-operated injectors designed for general utility in auxiliary mine-ventilation service, that is, for ventilating long dead ends through pipe lines. In accordance with the second group of laws, that apply to geometrically similar injectors equipped with geometrically similar nozzles operated at the same jet velocity, it is possible to show the performance of three sizes on one graph by merely changing the quantity scales.

The Probability Theory Applied to Exhaust System Design

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AMONG the possibilities to be considered in a study of the economics of exhaust systems is that of taking advantage of the probable non-use of all openings at one time. Although very few exhaust systems are operated so that all exhaust hoods are in use simultaneously, engineers are in many cases restricted by requirements or codes which provide for utilization of all openings.

An exception is that of exhaust systems for woodworking shops, where in some States the main branch may be designed on the assumption that not all openings are in use at any one time. In such cases, the main branch is designed so that the volume of air flowing in it is such that no material settles. A "50% system" design, therefore, provides higher air velocities when more than half of the openings are in use.

On the other hand, while regulations restrict the design of systems based on partial use of openings, they do permit blast or shut-off gates so that openings not being used may be closed. In some respects, this practice can be criticized to a greater extent than a "partial" system, since cutting off the desired air flow reduces the critical air velocity and this may lead to a deposit of collected material in the main branch.

This article outlines a statistical approach to the design of a partial system. It is of interest to engineers and perhaps may aid regulatory bodies to a rational approach to the interpretation of laws relating to the use of partial systems. The method is dependent on a number of assumptions, most important of which is that the operation of every opening is *independent* of the operation of every other opening. In other words, the method here proposed does not apply if a group of openings must always function simultaneously. In such instances the group must be considered as a single opening.

In addition to the above assumption, we must have prior knowledge of the following: (a) Average time of operation of each opening during a stated period, and (b), the maximum permissible period that a design number m of the openings will be in operation simultaneously. The design number will be a matter of computation along the line developed in the paragraphs which follow. It is essentially the answer to the problem of determining the probable number of openings in simultaneous operation for a stated percentage of time.

The problem as stated can be solved by the methods of probabilities provided, as already stated, that the operation of one opening is not contingent on the simultaneous operation of any other opening. Let t be the average time of operation of each opening of a system of n openings in a period of time T. Let t be measured in minutes and let T represent a period during which all the openings during one time or another have been in use. The period T may well be a day of 8 hours in some cases. In general, we may write T_r as the permissible period for r openings to be in operation simultaneously without fear that the collected material will clog the duct.

All things being equal the ratio t/T represents the probability of one of n openings being in operation at a given instant. The probability that it will not be in operation is then $(1 - t/T)$ or $(T - t)/T$. The probability that none of the remaining $(n - 1)$ openings will not be in operation is by the law of probabilities $[(T - t)/T]^{n-1}$ and the probability P_1^n that one of n openings will be in operation at a given instant is

$$P_1^n = n\left(\frac{t}{T}\right)\left(\frac{T-t}{T}\right)n-1$$

In general, if r openings are in operation at a given instant, the probability of such occurrence is

$$P_r^n = \left(\frac{n!}{r!\,(n-r)!}\right)\left(\frac{t}{T}\right)^r\left(\frac{T-t}{T}\right)^{n-r} = C_r^n \frac{t^r\,(T-t)^{n-r}}{T^n} \quad (1)$$

where C_r^n is equivalent to the term in the first parenthesis and represents the combination of n openings taken r at a time. If we multiply through by T_r

$$T_r P_r^n = T_r C_r^n \left(\frac{t}{T}\right)^r\left(\frac{T-t}{T}\right)^{n-r}$$

and set this equal to unity

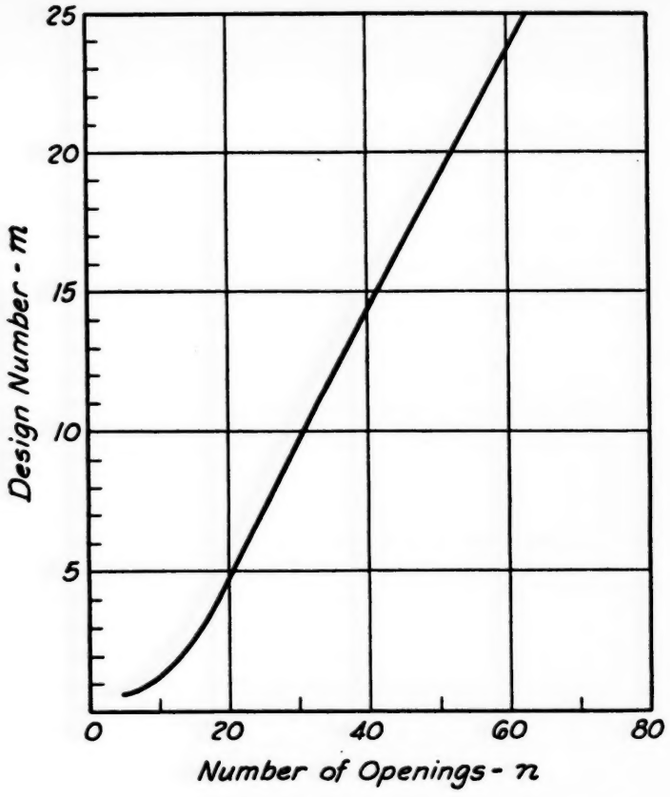
$$\frac{1}{T_r} = C_r^n\left(\frac{t}{T}\right)^r\left(\frac{T-t}{T}\right) \quad (2)$$

and we have the condition that r openings out of n will be in operation simultaneously one minute out of T_r minutes. If $r = m$, where m is the design number, we may say that this number will not be exceeded by more than a given fraction of the time $1/T_m$ given by the expression

$$P_r^n = \frac{1}{T_m}, \quad (r = m) \quad (3)$$

By trial and error, we may deduce from this equation the design number m.

We see from the above equations that we must have reasonably representative estimates of t, T and T_m. The value of t may be taken as the average time of



Relation between n and m for $t/T = ½$ and $T_m = 0.02$

operation of each opening. The time T is arbitrary, though it should encompass the time all units have at one moment or another been operating. Often some openings may not be used for days. It is best to assume operation of such openings for a few minutes time so that the interval T will not be too long. The value of T_m is more difficult to decide. It seems best to select a value of T_m sufficiently small so that the design number is not exceeded for too long a time. For most wood-working shops a value of 2% ($T_m = 0.02$) appears adequate.

In the example which follows we consider conditions common to a wood-working shop and assume during a half-day's working time (240 minutes) that a given machine will operate for half that period so that $t/T = 120/240 = ½$, in which case equation (1) becomes

$$P_r^n = C_r^n \left(\frac{t}{T}\right) = C_r^n \left(\frac{1}{2}\right)^n \qquad (4)$$

Under these conditions, using this equation and equation (3) with $T_m = 0.02$, we have the results shown in Fig. 1. These computations made by the methods of trial and error are tedious but unfortunately cannot be simplified.

The results shown in Fig. 1 are practically linear for large values of n. We see, however, that although each machine may be working half of the time, not more than roughly a third of them may be expected to be in operation simultaneously 2% of the time. We must, of course, appreciate that random operation of each machine is assumed. Otherwise the analysis fails.

If we know the average flow into each opening, we can determine the size of the main branch needed to fit the above conditions. Let us propose the following problem:

Given an exhaust system of 60-openings. The average volume of flow through each opening is 200 cfm. Except for 5 openings which are always operated simultaneously, the openings are used at random. Determine the design number of openings which will be in operation not more than 2% of the time if on the average each opening operates for 120 out of 240 minutes. Determine the size of fan entrance if the air velocity required for conveying the collected material is 2200 fpm.

1. The total volume of air handled is $60 \times 200 = 12000$ cfm.

2. The average volume handled per opening is $12000/56 = 214$ cfm. (Considering the five openings operated simultaneously as one opening).

3. The conditions of this problem are identical to those of the example in the text above. Hence, using Fig. 1, we have for 56 openings a design number of 22. Thus, we may expect that not more than 22 openings will be in operation simultaneously 2% of the time.

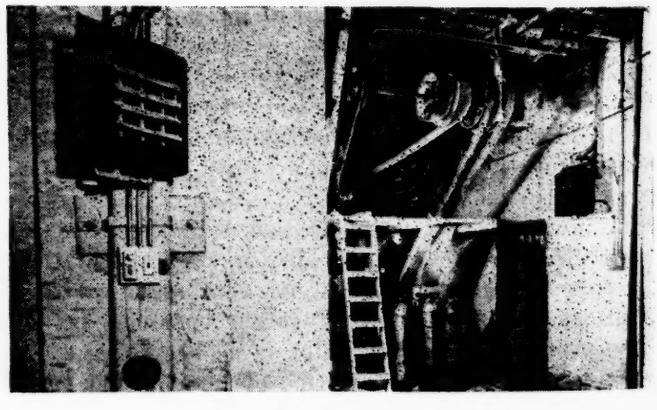
4. The design volume is therefore $22 \times 214 = 4708$ cfm.

5. The fan entrance area is $4708/2200 = 2.14$ sq ft, which is approximately equivalent to a 17 in. diameter opening.

We must remember that in the design of any partial system all openings not in use must be shut off. The above example illustrates that the use of a 50% system for wood-working as provided in several states gives a reasonable factor of safety.

ALL-ELECTRIC UNIT HEATERS

Mountain City Mill Co. became the Tennessee area's largest single user of electricity for space heating when 50 Electromode unit heaters were installed. The connected load for the mill's 598,650 cu ft is 655.5 kw, with heater units ranging in capacity from 7.5 to 20 kw. It was found that this load could be operated for a large part of the time off the peak of the 900 kw demand of the mill's productive equipment. Company officials say the first full season showed an appreciable saving in plant heating costs.



Chattel Mortgage and Real Estate Law

for Engineers and Contractors

LEO T. PARKER
Cincinnati, Ohio

Since better understanding of contract law results in fewer law suits, a review of important court cases is given to demonstrate the rights of engineers and contractors. Facts are given to avoid legal pitfalls.

LEGALLY, a chattel mortgage is a means for which a person who sells personal property, such as heating equipment, retains the title to the same until the buyer fulfills his agreement. Or the owner of personal property may mortgage it to obtain money or other services.

The mortgagor is the person who owns the property, and the mortgagee is the other person who may loan money on the property, or the former may be the seller of property on which he holds a mortgage as security for the balance due on the purchase price.

Since the law pertaining to chattel mortgages is commonly involved in the sale of heating equipment, we shall endeavor to explain the important phases of the law based on higher court decisions decided recently. For example, an important question is: When is an unrecorded mortgage on heating equipment valid and enforceable?

In the case of Clover Specialties Company, 18 F. (2d) 314, the United States Court considered the effect of an unrecorded chattel mortgage. This court held that an unrecorded chattel mortgage is not effective against persons who do not know of the existence of the mortgage, but it is valid and enforceable against all persons who are familiar with the fact that the mortgage exists.

And in U. S. Line v. Colburns, 255 Pa. 688, the court held that after a chattel mortgage is properly recorded it is effective against all persons irrespective of whether or not such persons have actual notice that the mortgage exists, because the public records are available showing the status of the mortgage. Persons transacting business are duty bound to search the mortgage records in all counties in all states to ascertain if the property involved is encumbered.

For comparison see Carter Guaranty Company v. Cumberland, 292 S. W. 812. Here it was disclosed that a mortgagor mailed a mortgage to a county clerk with a check for the cost of recording it. The county clerk cashed the check but neglected to record the mortgage. This court held the mortgage not recorded, and explained the law, as follows:

"The only proof that the mortgage was so received is that it was inclosed in the envelope with the check and letter of instructions and the check was returned to the drawer with the endorsement of the clerk thereon. . . . The question is whether the return of the check indorsed by the county clerk is proof that the mortgage was delivered to the county clerk, and, if it should be held that it is proof of delivery . . . without proof that the mortgage was delivered to the clerk or some one having authority to receive it in the office of the clerk and the fees paid thereon, it was not on record."

As previously stated, an unrecorded mortgage is valid against all persons who know that the mortgage exists. For illustration, in the recent case of Wertheimer & Degen v. Schultice, 211 N. W. 569, the court rendered a decision with respect to the validity of chattel mortgages, in which it was held that a chattel mortgage is good against all persons who know of its existence, irrespective whether or not it is recorded in accordance with the law.

And in Union State Bond v. Warner et al, 248 Pa. 395, the higher court held that a chattel mortgage is valid and effective where it is recorded and unknown to the complaining party.

Sometimes a person becomes involved in litigation when he buys a building not knowing that the heating equipment is mortgaged.

In the case of Messenger Publishing Co. v. Overstreet, 137 S. E. 125, the court explained the law on this subject and said:

"Where one purchases mortgaged property with actual notice of the existence of a certain mortgage thereon, the description of the property in the mortgage need not be of a definiteness sufficient to constitute constructive notice by a record of the mortgage to a purchaser of the property, but need *only be of a definiteness sufficient to constitute a good description as between the parties to the mortgage.*"

Therefore, it is quite apparent that the description of the mortgaged property in the mortgage may be vague to a person unfamiliar with the property, and yet the mortgage is valid if the description is clear to the mortgagor and mortgagee.

The recent case of Security State Bank v. Jones Mercantile Company, 247 Pac. 862, involved the description of mortgaged property. This court said:

"The general rule applicable to a description of property in a chattel mortgage is that if the description given is such as will enable third persons, aided by reasonable inquiries . . . to identify the property, it will impart notice and be sufficient. . . . Although the description may be meager and even defective, if it fairly gives a clue to the identity of the property so that third parties may by reasonable investigation ascertain the property which the parties intended to include in the mortgage, the instrument must be regarded as a valid lien. The location of the property

at the time the mortgage is executed affords a clue by which the property may be identified."

Lease Contract Law

So many times heating and ventilating contractors install equipment in a leased building and have difficulty in collecting the contract price. So, therefore, for the benefit of contractors who may encounter legal difficulties involving lessees we shall briefly review this law.

Probably one of the most important points of the law relating to leases is this: While an ordinary verbal agreement relating to the sale of chattels is equally valid and binding as a written contract, yet the law requires agreements relating to real estate to be in writing. In other words, neither party can compel the other to fulfill the terms of an oral real estate agreement unless, of course, the complaining party has expended money relying upon the other's promises.

The latest case on this subject is Forbes v. City of Los Angeles, 282 Pac. 528. Here it was disclosed that a property owner made a verbal contract by the terms of which he agreed to sell his property. Later the property owner filed suit to rescind the agreement on the contention that the contract was void because it was not in writing. It is important to know that the higher court held the oral contract void, stating the following important law:

"It was necessary to prove either that the contract was in writing or that it had been partially performed. . . . The case is reduced to an action to enforce a parol contract to convey real estate. Such an action is maintainable *only where there has been a part performance sufficient to take the agreement out of the statute of frauds.*"

Purpose of Real Estate Laws

The purpose of the laws, which require real estate sale and lease contracts to be in writing, is that the assumed obligations of the parties may be unmistakenly recorded. Therefore, testimony relating to correspondence of conversation between the parties, occurring prior to signing the lease contract, ordinarily is not admissable in later suits involving the contents of the lease. However, any evidence is admissable in litigations involving leases, if such testimony relates to deceit, fraud, or misrepresentation of a material fact made by one party for the purpose of inducing the other to make the contract.

For instance, in Holloway v. Pegran, 71 S. E. 367, it was shown that a series of preliminary letters passed between the parties previous to the execution of the lease contract. In these letters the landlord made certain fraudulent promises by which the prospective lessee or tenant was induced to enter into the lease contract. Later the tenant sued the landlord to cancel the lease.

Under these circumstances, since the testimony proved that the landlord had mailed the letters expressly for the purpose of fraudulently inducing the tenant to make the lease contract, such letters were admissable as evidence to prove deceit, and the lease was void.

Agreements Subsequent to Contract

On the other hand, it is well settled that any written memorandum signed by the contracting parties and attached to the lease contract, becomes a valid addition to the original agreement.

For example, in the recent case of Illinois Co. v. Beifeld, 184 Ill. App. 582, the litigation involved the validity of a memorandum which was signed by the landlord and tenant and attached to the original lease several months after execution of the lease contract.

The court held the memorandum to be a part of the contract and valid, because it was signed by the parties and attached to the contract. The same law is true with respect to other written agreements which, although not attached to the lease contract, are deemed to be related to the contract, but not a part of the lease itself.

For illustration, in another leading case (203 N. Y. S. 648), a property owner, who contemplated leasing his property for forty years, wrote a letter to a prospective tenant in which he agreed to grant a renewal of the lease for an additional forty years. The lease contract subsequently executed contained no provision for a renewal of the lease.

At the expiration of the first forty years, the landlord refused to grant the renewal on the contention that the renewal clause was not included in the lease contract. However, the court held the tenant entitled to a renewal, saying:

"The law is well settled that an agreement in writing to give a lease, which is clear and explicit, may be enforced as a contract between the parties, although a formal lease may not have thereafter been executed."

When Termination of Lease is Valid

It is commonplace for lease contracts to contain provisions which confer upon the property owner an option to terminate the lease before the expiration of the term. It is important to know that no particular form of words is necessary to confer this right. However, certain words, namely, "upon condition," "provided," "reserving," "subject to," or their Latin equivalents, are recognized as customary and valid. On the other hand, it cannot be presumed that a landlord, or lessor, has the privilege of terminating a lease unless the contract clearly indicates that such was contemplated by the parties when the agreement was signed. Obviously, therefore, a landlord cannot terminate a lease contract by performing a breach of the contract and then endeavor to introduce proof of this act to induce a jury to believe that the tenant intended termination of his lease under these circumstances.

A lease of property may be cancelled by the surrender of the leased premises and the acquiescence in such surrender by the owner. Such acquiescence is perhaps best evidenced where it is shown that the owner took possession of the property and assumed again all the authority over it.

Under the law, a surrender of a lease can be made only by express consent of the parties, in writing, or by operation of law, as when the parties do something that implies that both have consented to the termina-

tion of the lease. Also, the law is well settled that a mere statement by the property owner to the lessee that the latter may move out if he desires, does not terminate the leasehold until *a surrender and its acceptance* have *actually* been made. Moreover, the act of the landlord in attempting to secure a new tenant for the premises, which the lessee has vacated, is not a legal acceptance by the owner of the surrender by the lessee. These various phases of the law were discussed in the recent case of Borchers Bros. Co. v. Ciaparro, 287, Pac. 113.

In this case the court said:

"A surrender of a lease is the yielding up of an estate. . . . Under the statute of frauds it can be done only by express consent of the parties in writing, or by operation of law when the parties do something which implies that both have consented. . . ."

When Tenant is Bound to Make Repairs

The majority of lease contracts contain clauses obligating the lessee or landlord to maintain the premises in good condition and repair. Therefore, it is interesting to observe that the courts have consistently held that clauses of this nature are construed in favor of the tenant. If the landlord fails to make necessary repairs specified in the contract, the tenant may pay for such repairs and deduct the expenditure from the rent.

For illustration, in Monnig v. Easton Co., 27 S. W. (2d) 495, it was shown that a lease contract provided that the landlord agreed to make necessary repairs to the heating system. The tenant complained about the poor condition of the equipment, but the landlord failed to remedy the condition. The tenant proceeded to remedy it at a considerable cost and deducted the amount from the rent. The landlord sued, but the higher court rendered a decision in favor of the tenant.

Lease Must Have Legal Objective

Another well established point of law is that if the object or purpose of a lease contract is illegal, neither the property owner nor the lessee is liable in damages to the other for breaching the contract. On the other hand, if one party performs an illegal act by which the other party is prevented from carrying out the terms of a legal lease, the former is liable in damages.

For illustration, in New Chester v. Bischoff, 206 N. Y. S. 641, it was disclosed that a tenant entered into a contract with the owner of property by which the latter agreed to construct a building according to certain plans and specifications, and lease it to the former for a term of fifteen years.

After the building was ready for occupancy and the two parties had signed the lease contract, the lessee was notified by the building inspector that the building did not conform with the requirements of the building laws, and he was not permitted to conduct the business.

The lessee sued the landlord for the damages and in holding the landlord liable, the court said:

"Had the plaintiff (lessee) known that the building was not constructed in accordance with the requirements, he could have refused to have taken possession. Having entered into possession without knowledge of the fact, the plaintiff (lessee) had the right to surrender possession as he did, and to recover the deposit and any damages he had sustained."

The Law of Repairs

Generally speaking, any reasonable provision in a lease contract is valid and enforceable which obligates either a lessee or the landlord to keep the heating equipment in repair. Moreover, failure of the obligated party to fulfill the terms of the agreement may result in his liability.

However, it is equally important to know that the owner of a building is not liable for damages as a result of defects in the leased building, where the lessee was aware of the defects, and failed to notify the landlord, irrespective whether the lease contract obligates the latter to keep the building in repairs.

Moreover, while it is the legal duty of a lessee to notify the landlord of necessary repairs, if the landlord fails to make the same within a reasonable time, it is the further duty of the tenant to make and pay for the repairs and charge the cost of the same to the landlord.

In other words, where a lease contract clearly obligates the landlord to keep the premises in repair, the tenant may not stand by and permit damage to result, although he notifies the landlord of needed repairs and the latter neglects to perform the required work.

Obviously, however, the landlord is liable if the lessee or tenant sends notification that the premises need repairing and, without negligence of the tenant, the defective condition of the premises results in damage to the tenant.

Plastic Piping

The story of how world-wide routes affected the choice of piping material for the 80-passenger Boeing Strato cruiser was revealed when Chief Engineer Edward C. Wells announced that plastic water pipes—435 feet of them—would be used throughout the new four-engine transport. Studies have shown that the difference in the mineral content of water throughout the world sets up chemical reactions in metal pipes. Extensive testing on certain types of plastic tubing, however, has indicated negligible chemical reaction to all types of water. When installed, sections of the Tennessee Eastman Co.'s Tenite are joined by short lengths of polyvinyl chloride. The joint is secured by two light metal clamps. The majority of valves are annodized aluminum alloy, given an organic finish to protect the metal against chemical reactions.

Acoustical Linings for Ducts

FRANCIS A. WESTBROOK

FREQUENTLY some space that is normally noisy, is sound conditioned with the result that while the original sound is abated, the noise emanating from the fan of the ventilating or air conditioning system, becomes audible through the ducts. In other cases private offices especially designed for privacy actually have no such thing because a duct system acts as a speaking tube. As a result of investigations and studies, it has been found that these difficulties can usually be overcome by installing the right kind of sound absorbing material in the ducts and in the right way to meet the given set of conditions.

Acoustical Duct Lining

Acoustical materials for absorbing air transmitted sound in duct systems should be moisture resistant because when used as duct linings they may be subjected to considerable humidity, especially when the ducts are part of an air conditioning installation. Fire resistance is highly desirable.

Low thermal conductivity is another essential characteristic for acoustical duct lining so that insulation on the outside of the duct is not necessary.

Sources of Duct Noise

There are four general noise sources encountered in ducts—grilles, air turbulence, cross talk, and fan, Fig. 1. The first is a high pitched or rushing noise due to the passing of air through wall grilles. This sound intensity depends on the design of the grille and the air velocity. As a rule this is negligible for air velocities usually employed in well designed systems. However, if there should be trouble from this source, the size of the grille may be increased to decrease the air velocity passing through, or by using a grille of a different design. In other words, noise from this source can usually be corrected without the use of acoustical material.

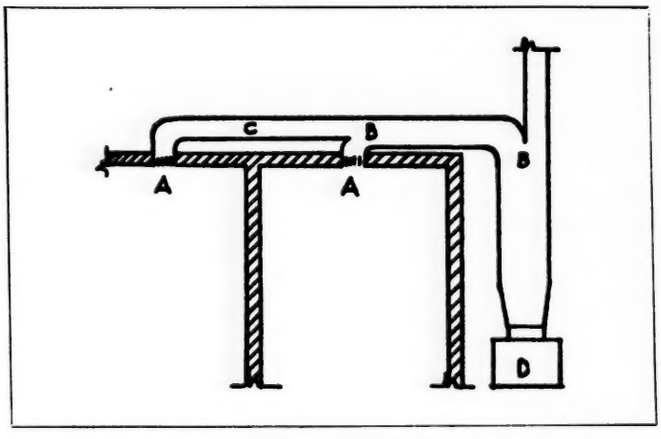


Fig. 1. Sources of ventilating noise; (A) grilles; (B) air turbulence; (C) cross talk; (D) fan.

Turbulence in the ducts is caused by obstructions in the duct system. As this sound increases with air velocity, the principal source of trouble is in the large ducts near the blower where the highest velocities occur. Control of turbulence by proper design of elbows, branches, dampers, splitters and other details directing the flow of air will do much to quiet the noise or to reduce it to a satisfactory extent. If it is not possible by careful design to control successfully the noise due to turbulence, it may be reduced to the extent desired by installing acoustical duct lining between the outlets and the points where the noise originates.

The transmission of sound, such as talking or music, from one room to another through a common duct is the third source of noise. It is of major importance in schools, radio studios, private offices and consulting rooms. It may be controlled by making the path between the rooms by way of the duct as long as possible or by lining the ducts connecting such rooms with acoustical materials.

Fans, in general, are the major offenders. The size and type of fan, and the speed at which it is operated, are the principal factors determining the amount of noise introduced into the duct system. To avoid unnecessary noise from this source it is important to install a fan of proper capacity for the duct system it serves and to operate it at its most efficient speed. Operation at higher speeds results in excessive turbulence and consequently unnecessary noise.

Amount of Noise Reduction

The object of controlling duct noise is to hold it down to a level where it is either inaudible in a given room, or it does not add noticeably to the normal noise level with the duct system not in operation. Thus the amount of noise reduction necessary is determined by the quietness of the room without ventilation or air conditioning, and by the noise transmitted by the untreated ducts.

There are too many variables for exact recommendations for all cases but an approximate guide has been formulated for average requirements, as Table 1.

Table 1 — Recommended Reductions of Ventilation Noise

Type of Room	Reduction in Decibels
Studios, Residences	30
Hospitals, Auditoriums, Private Offices	20
Quiet General Offices, Quiet Restaurants	10

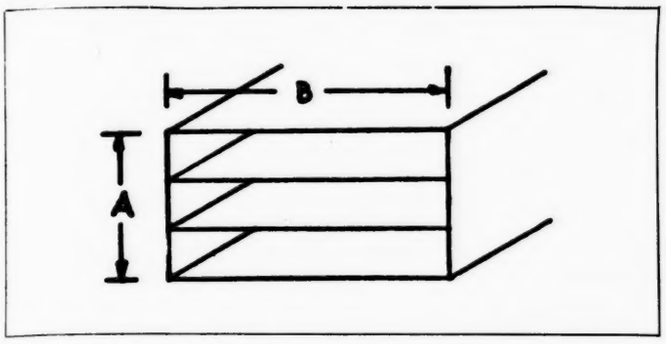


Fig. 2. Use of splitters to increase quieting.

This indicates very clearly that in normally noisy places, such as offices and restaurants, less reduction of duct noise is required than in quiet places like residences and studios. In the case of cross-talk between rooms, the reduction must be increased by installing enough duct lining to equal the reduction furnished by the walls separating the rooms.

Calculating Noise Reduction

Noise reduction, in decibels, of sound travelling inside a duct treated with acoustical material has been shown by experiment to be directly proportional to the length so treated and to the ratio of the perimeter of the duct to its area of cross-section. Therefore the greater the length of lined duct and the smaller it is, the greater is the reduction of noise. The reduction also is proportional to about the 1.4 power of the absorption coefficient of the acoustical material. These facts are expressed by the formula, accurate to within plus or minus 10% for coefficients between 0.20 and 0.80:

$$R = 12.6\, L \frac{P}{A} a^{1.4}$$

where R = reduction in decibels,
L = length of lined duct, feet,
P = duct perimeter, inches,
A = area of cross-section, square inches,
a = absorption coefficient of acoustical duct lining.

Several facts of practical value are indicated by this formula. For example, the smaller the duct the larger is the ratio P : A, and consequently the more effective each linear foot of acoustical lining. Each linear foot of lining is less effective in a square or circular duct than in a rectangular cross-section of the same area. A square foot of lining produces the same noise reduction in ducts of the same cross-sectional area, regardless of their shape. This means that similar amounts of material are required to quiet ducts of similar capacity, but the length of duct to be lined depends on the shape.

Effect of Frequency

As the absorption coefficient of a given acoustical material varies with the frequency of the sound, the reduction of noise in a treated duct depends on the frequency. Table 2 gives the absorption coefficients of ½ and 1 in. thick material (Q-T Ductliner made by The Celotex Corp.), plus the corresponding decibel reduction formula for each frequency. As an example of how this is used consider a duct 12 in. × 12 in. × 12 ft. long lined with 1 in. thick material. The value of P/A is 48/144, or 1/3. Reference to Table 2 shows the reduction for a noise of 256 cycles frequency to be 3.6 × 1/3 = 21.6 decibels. If the sound has a frequency of 2,048 cycles the reduction in the same duct is 10.5 × 18 × 1/3 = 63.0 decibels.

Table 2 – Decibel Reduction Formulas at Single Frequencies

Freq.	½ in. Q-T Lining		1 in. Q-T Lining	
	Abs. Coef.	Decibel Reduction $\frac{P}{A}$ in L —	Abs. Coef.	Decibel Reduction $\frac{P}{A}$ in L —
128	0.14	0.8	0.29	2.2
256	0.38	3.3	0.41	3.6
512	0.43	3.9	0.78	8.9
1024	0.76	8.6	0.89	10.7
2048	0.75	8.4	0.88	10.5
4096	0.75	8.4	0.78	8.9

Sound transmitted through a duct generally contains a large number of frequencies covering most of the range of audibility. The overall reduction in an acoustically treated duct depends on the frequency distribution and the reduction of those frequencies for which the absorbent material is least effective. Analyses of noises emanating from fans have shown that the loudest part comes from the frequencies around 256 cycles. Consequently in calculating the overall reductions for fan noises in decibels, the formula in Table 2 for 256 cycles should be used.

If the noise to be abated comes from air turbulence, and not from the fan, values for 1,024 cycles should be used. Most of the energy in noise of this type is in the higher frequencies, centering around 1,024 cycles. Where cross-talk between rooms is to be handled, reductions on the basis of 256 cycles should be selected as this will include all types of sound likely to be involved.

Use of Splitters

Often the length of duct, where lining is to be installed, is too short to permit the required reduction, and in such cases it must be divided into small chan-

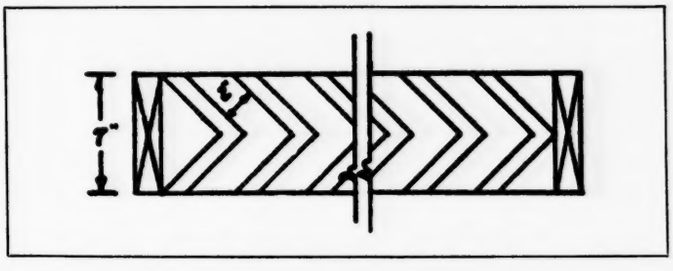


Fig. 3. Construction of baffles.

nels by splitters made of the duct lining material. These splitters must be installed parallel to the widest side of the duct as shown in Fig. 2, with both sides of the acoustical material exposed, and the walls of the duct lined in the usual manner. A formula for calculating noise reduction by means of splitters is

$$R_s = R_o \frac{A + BN}{A + B},$$

where R_o = reduction in decibels in lined duct without splitters,
R_s = reduction in decibels in same length with splitters,
A = length of short side in inches or feet,
B = length of long side in inches or feet,
N = number of channels formed by the splitters.

Consider, for example, a duct 18 by 24 inches in cross-section and long enough to permit of only a 10 decibel reduction by lining the four sides, although conditions call for a reduction of 30 decibels. By installing four splitters parallel to the 24-in. side five channels are made and the formula for the reduction will then be

$$R_s = 10 \times \frac{18 + (24 \times 5)}{18 + 24} = 32.9 \text{ decibels}$$

Use of Baffles

If the duct space is even more limited, so that the desired reduction cannot be obtained with splitters, effective noise reductions can still be obtained by the use of baffles made of the duct lining material. These are given a zig-zag arrangement in the path of the sound waves, as indicated in Fig. 3, but offer only a comparatively small resistance to the flow of air. Tests made on an assembly of four baffle units in tandem consisting of 1 in. thick material, with both sides exposed and installed in a space occupying only 28 in. along the duct gave the following results:

Frequency	128	256	512	1,024	2,048	4,096
Sound reduction, decibels	6	7	15	34	47	48

Some efficiency of air flow is, of course, necessarily sacrificed and unless this is permissible the method will not be practicable.

Decibel System

The decibel system of measuring the quantitative effect of sound provides a means of measuring the reaction of the ear to noises of different intensities or loudness. Loudness as a sensation follows a logarithmic law with respect to physical intensity. The "threshold of hearing" is at 0 decibels and the "threshold of feeling," where the loudness begins to become painful, is at 120 to 130 decibels. The physical intensity, or energy of the sound waves at 130 decibels is some 10 trillion times greater than at 0 decibels.

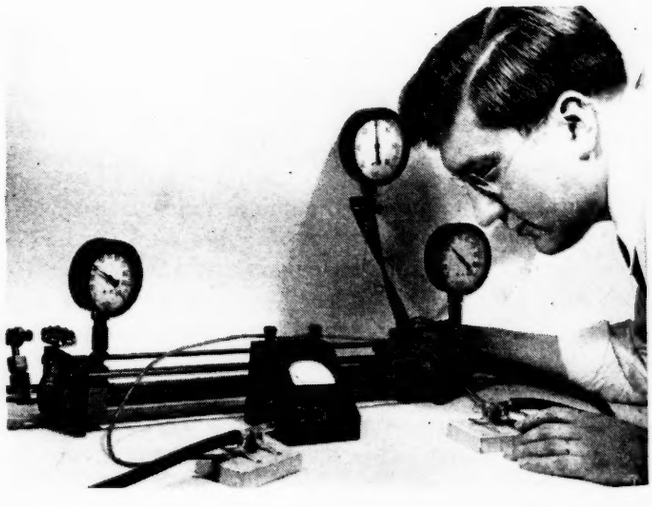
Decibel values of some of the usual sounds are: whispering, 20 to 35; average conversation, 55 to 65; noisy office or average factory, 60 to 80; noisy factory 80 to 100; riveting, grinding, airplane engines, etc., 95 to 140. The point where noise becomes dangerous to health begins at around 90 decibels, but it varies with the individual, and even if actual pain is not felt the effects of long exposure to noise around that level may have bad physiological results.

Scientists Study Hot-Cold Pipe

A "hot-cold pipe" 15 in. long and 1 in. in diameter that converts ordinary compressed air into both hot and cold air without the aid of moving parts has been built by Westinghouse Research Engineers. As yet scientists have no explanation for the phenomenon though it is conjectured that the cause may be a frictional effect between gases moving at different velocities. To study the principle involved a working model has been made.

In operation compressed air is pumped into a nozzle at one end of the pipe. There it strikes the inner wall of the pipe tangential to the diameter and is converted into a whirlpool of spinning gases. The air in the center becomes cold almost instantly while that on the periphery gets warm. The cold air is drawn off through a ¼-in. opening at the center of one end, while the warm air is tapped from the periphery.

With the original tube .3 in. in diameter, R. Hilsch, its German inventor, claims to have produced simultaneous jets at 154F and 10F. The present model has purposely been constructed larger in diameter to expedite instrumentation for scientific study. For this reason it gives a temperature difference of only several degrees F.



Low efficiency rules out commercial application at this time, but like the "blow glow" in Edison's experimental electric light that led to his invention of the first electronic tube, it may lead to unexpected developments. Even now it holds promise as a handy laboratory tool for making cold as simply as a Bunsen burner makes heat.

Experience is my Classroom

By T. W. REYNOLDS

KEEP YOUR SHIRT ON

The man who keeps his shirt on in a very hot area will be cooler to a greater extent than if he were half naked, because the shirt or undershirt provide an increased amount of evaporating surface. It also acts as a partial shield from high temperature radiation. You can ventilate where there is radiation from hot surfaces and as a general rule do so ineffectively, for the air will not eliminate the radiation, but the moving air will evaporate moisture from the shirt and so will the radiation.

Drinking plenty of water will help to cool the body by increasing the amount of sweating. The salt lost with the sweat should be replaced by adding salt to the drinking water. Of course, working in a drenched shirt is irritating to the body, increases laundry bills, and is generally distasteful to most workers.

A certain draftsman worked in his shirt sleeves in the winter time and wore his coat in the hot summer months,—said he wanted to shut the heat away from his body in the summer. Over in Russia it is said that they do it differently—that is, wear their outdoor clothes when inside, so that they carry the stored heat of their clothing out with them when going outdoors. Anyone who keeps his overcoat in an unheated closet can appreciate this.

PRE-VENTILATION

Quite often we precool or preheat a space prior to occupancy by closing the outdoor air intake dampers, meanwhile recirculating all of the air. Certain spaces, however, such as wash rooms, rest rooms, locker rooms and toilet rooms, require a reverse procedure, something seldom done. These spaces require a thorough airing out before being used, leaving doors and windows open for the exhaust fan to pull fresh air through. This will remove the overnight odors as absorbed by floor and other surfaces.

Note: Mr. Reynolds is Chief, Air Conditioning Division, Abbott, Merkt & Co., Engineers, New York.

JOINTS IN BUILDINGS

Expansion joints are a necessity in skyscrapers, but the need for an access door at each joint is open to question, considering the expenditure. In lieu of a door, the plaster can be left off to make an observation hole opposite the joint until the building is about complete. Patching men can then close up the openings.

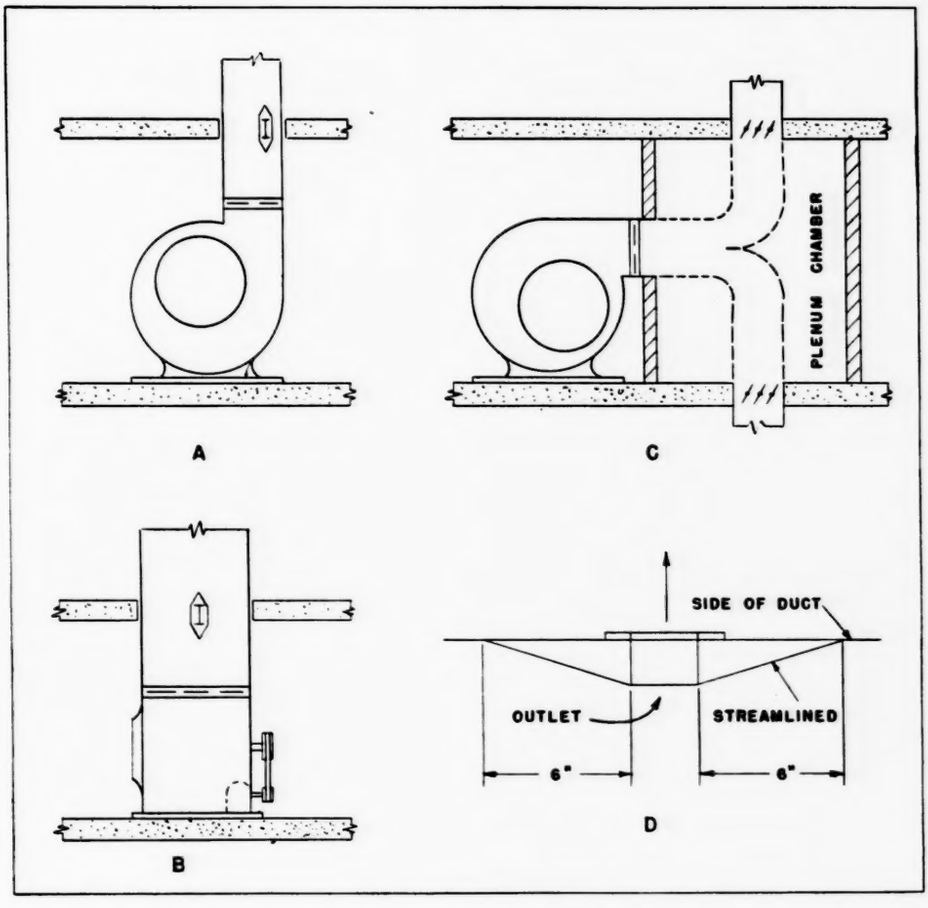
A joint may develop trouble, possibly due to a pin hole in the spinning process. Where they do, the wire lath should be cut to prevent plastering over until the trouble has been corrected. If the lath is not cut, the trouble may be forgotten; meanwhile someone may close it up as a matter of routine. When runouts are in the floor fill, they can have but slight movement. In such cases, the location of expansion joints must be given special consideration. Chases in walls for runouts should be made big enough to allow for expansion.

STREAMLINING IN DUCTS

When there is a beam in the way of a fan discharge, the first thing that is done is to streamline it. This is all very well, depending on just how it is done. If the arrangement is made as shown in *A* (drawing below), there will be trouble, because the beam blocks the major point of the air discharge which hugs the scroll of the fan. Turning the fan around 90 degrees, as shown in *B*, avoids this difficulty.

A fan arranged as in *C* so that its discharge splits almost immediately into two ducts, will also give trouble. The air hugging the scroll of the fan will favor the upper duct. This situation can be remedied by making a plenum chamber for the fan discharge, by eliminating the two ducts within the chamber as shown dotted, and by adding a louver damper to each duct inlet.

Sometimes it is necessary to place a supply outlet directly in the duct work. In such a case a streamlined baffle 6 in. long should be placed on either side of the outlet, as in *D*.



Streamlining in Ducts

OLD HOTEL MUST GO

Back in 1884 the barns of the Madison Ave. bus line at 40th St. and Park Ave., New York, were torn down to make way for the last word of the times in the way of a hotel known as the Murray Hill Hotel. Now, in 1947, this old familiar hotel will in its turn be replaced, this time by a 30-story modern office building. The newspapers of those days were lavish in their praise of the new hotel with its 8 stories and 575 rooms to cost in all about 1½ million dollars. The hotel was variously described as one of the finest and most magnificent in the city, the talk of the town, a veritable palace and a most sumptuous hotel in all respects. Here were elegance and refinement of equipment contained in wide halls and spacious rooms with high ceilings.

The hotel opened at $4.00 per day on the American Plan with many rooms engaged in advance. Ladies in tiers of petticoats graced its red plush chairs amid heavy plush draperies, gold pillars, a Victorian atmosphere and a lot of chirping canaries. Around the hotel exterior hung heavy cornices and gargoyles sculptured in grotesque, allegorical and mythological subjects.

The newspapers dwelt on the growing preference for up town sites, the quiet and select surroundings of our up town avenues, the proximity of the hotel to the fashionable quarter, its convenience to the Grand Central Depot (saving cab hire) and the advantages of a location on the highest and healthiest ground on the island of Manhattan, adding that this hotel offered a good place for a permanent residence for families.

Neighboring hotels were listed as the Grand Union, Belmont, Manhattan and Knickerbocker, all of which have long since gone their respective ways. Only three blocks south on Park Ave. was the home of Mary Lindsley Murray who entertained General Howe and his troops at her home until the American troops under General Putnam escaped and so the hill became known as Murray Hill.

A tower or covered observatory at the north east corner of the building was described as a place 180 ft. from the ground where one may obtain a birds eye view of all Manhattan Island, together with the surrounding country and the blue mountains in the distance. In the lower portions of the tower were located the Maids' Rooms, now empty and with many of the windows gone. Visited by the writer but recently, progress up the stairway was impeded by countless pigeons whose wild flutterings expressed their amazement at such a rare visit to their holdings. These are the pigeons who forage for many blocks around, make their deposits at night, now a foot deep, and maintain the bird's eye view of an everchanging metropolis. Pigeons notwithstanding, the climb was completed to the top where one finds the last resting place of the old cuspidors and bed chambers once imported from England. These bits of crockery which once served mankind in many a moment of emergency and distress were long ago stored there and eventually forgotten.

The newspapers also commented fully on the 150 bath tubs, 300 lavatories and 120 Motts water closets as installed by the contractor Joseph P. Quinn of 251 Fourth Ave. Mr. Quinn was set forth as a plumber for 45 years, one who employs 25 gangs of workmen and who did the plumbing for the Hotel Windsor, Gilsey House, Park Ave. Hotel and the new Roman Catholic Cathedral. The waste pipes were said to be unusual in that they were back vented, each parlor had a fireplace, each room hot and cold water and there were Otis hydraulic elevators. The first two floors were lighted by the Edison incandescent light while the upper floors were lighted by gas. To quote the papers "A novel arrangement is to be used for lighting the gas by electricity, when the gas is turned on, a spark lights the gas."

The architect was Stephen D. Hatch of 115 Broadway with 25 years of experience in the design of many of the city's fine private mansions, as well as banks, such as the Union Dime Savings Bank. George Murray described as a member of Hawkins Zouaves during the Civil War and a steward of the Hotel Windsor, New York, became the steward at the Murray Hill hotel. The hotel was leased by Hammond & Hunting and placed under the personal direction of Mr. D. S. Hammond. The latter was identified as a member of the New York Stock Exchange, manager of the Hotel Bristol of New York and proprietor of the Palisades Mountain House at Englewood-on-the-Hudson.

The original owner of the hotel was Hugh Smyth and from his estate Benjamin L. M. Bates bought the hotel some years later for $1,796,000 through Joseph P. Day at an auction sale. Mr. Day is said to have offered Bates $6,500,000 for the hotel only ten years after the auction sale. Bates subsequently shared control of the hotel with J. Mc E. Bowman.

Many notables frequented the hotel or stopped in to visit the famous Horseshoe Bar, such as the elder J. P. Morgan for brandy, Mark Twain for a Bourbon Toddy, Diamond Jim, Jay Gould, P. T. Barnum, Grover Cleveland and President McKinley.

Many decisions by the OPA in favor of the old time tenants (about 300 residents) of the hotel have delayed the wrecking of the old hostelry. Many of these residents have been there 20 to 30 years while a few are said to be 40- to 50-year residents. A porter has worked there for 30 years, the chief engineer 15 years, and his father before him 20 years and one of the firemen has been there 28 years. The engine room contains the generating equipment, ancient but well maintained, all in a closely confined height and not far away are the old wine cellars.

TAGS WOULD REDUCE WASTE

We don't seem to grasp the significance of waste until it is termed "loss". A man can waste $100 worth of material and it may be overlooked as something that goes with a process, but let it be said that he had lost $100 worth of material and there would no doubt be an inquiry.

In a small shop so many dollars worth of material is apt to be valued as such, while in a large plant it is relatively of little consequence. A lone sack of cement, for example, has a value that is greatly depreciated in one's mind as soon as it is moved so as to form but a mere unit of a huge stock pile.

What savings there would be if each bit of material were stamped or tagged to show its cost! The lumps of coal so identified would be viewed in another light. The fireman who burns a large amount of coal just to make a small amount of steam would no doubt cut down on the coal used and increase the quantity of steam produced.

THE NEED FOR PLANS

One thing that a young engineer must learn and some older engineers should learn, is that you have to get something drawn up in order to get something done. If you wait for the many specialists of today and others that are concerned to first submit their drawings, you may wait indefinitely.

Others are governed somewhat in their design by what you do in yours and all designs should proceed simultaneously and in co-operation. The interior decorator may have a general plan which you cannot infringe upon, but he cannot supply details and know how to blend in with your equipment until you first set down what equipment and where located.

An executive office layout is one of those things that try men's souls. Changes are continuous and the final ones are usually only after partitions are all up. There are so many persons involved and no one seems to know just what they do want until they see something on a drawing which they know they do not want. Furthermore, when a drawing is before them, they are forced to reach some decision.

Layouts for kitchen equipment and industrial machinery are still further examples of things which arrive last and late. Frequent pipe outlets of liberal size can, however, be provided without much additional expense as a hedge against most probabilities.

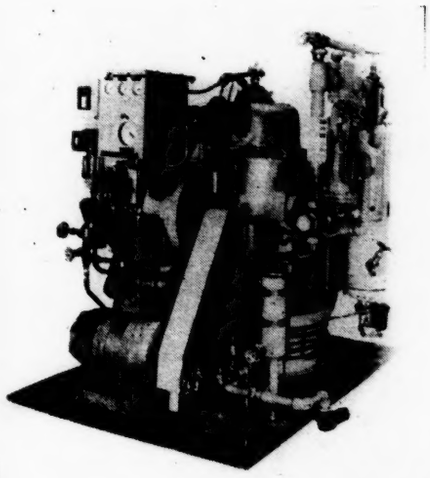
NEWS OF EQUIPMENT AND MATERIALS

Steam Generator

NAME—Vapor-Clarkson steam generator.

PURPOSE — For heating passenger trains powered by Diesel locomotives, and for general use in the heating and process steam field.

FEATURES — Generators are forced circulation, coiled water tube type, using oil as a fuel. From 300 to 3,000 lb of steam per hour are generated by heat exchangers ranging to 1,000,000 Btu per sq ft of heating surface. Generators are designed to



occupy a minimum amount of space and are self contained except for fuel and water supply.

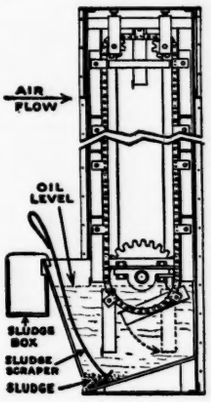
MADE BY—*Vapor Car Heating Co., Chicago, Ill.* **467**

Air Filter

NAME—Uniflo.

PURPOSE — Automatic self-cleaning air filter.

FEATURES—Unit consists of a steel cabinet. At the bottom is an oil tank and around the inside are filter cells suspended from chains Cells move up the front of the cabinet like a curtain, then across the top and down the back to pass through the cleaning tank at the bottom before again starting on the upward climb. Carrier chain is actuated by an electric motor with suitable gear reducers and is controlled by an electric time control. Filter medium consists of formed strips of aluminum or other metal with the corrugations in each adjacent element so placed that they run in opposite directions. This forms a honeycomb through which the air must pass. The dirty cells are held submerged for about 30 min and this breaks the surface tension of the liquid which binds the accumulated dust to the filter medium.



LITERATURE AVAILABLE—Illustrated folder.

MADE BY—*Continental Air Filters, Inc., Louisville, Ky.* **468**

Air Filter

NAME—Trion electric air filter.

PURPOSE—Air cleaning in the home through electrostatic precipitation.

FEATURES—Inside cabinet are 59

parallel aluminum plates which form a collecting chamber, and an ionizing screen consisting of six fine wires suspended between aluminum tubes. A power pack with transformer and tubes supplies high voltage direct current to the screen and collecting chamber. A jet spray system washes the collected dirt down the sewer. Controls consist of a power switch and a washing valve. They are mechanically and electrically interlocked for absolute safety and convenience.

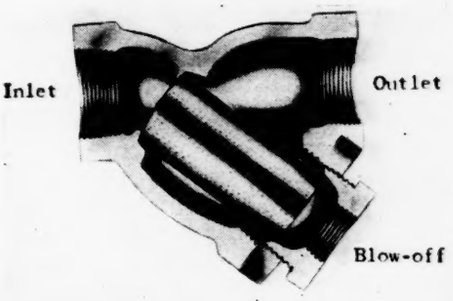
SIZES AND CAPACITIES—Model 100 for homes up to seven rooms and Model 200 for homes up to 11 rooms.

MADE BY—*Trion, Inc., 1000 Island Ave., McKees Rocks, Pa.* **469**

Pipe Line Strainer

NAME—Wright-Austin Y-type strainer.

PURPOSE—To prevent fine foreign matter from passing into steam



traps and other sensitive pipe line equipment.

FEATURES—Strainers are self cleaning by opening a blow-off cock which flushes grit and dirt out of the strainer by line pressure without shutting off the line or removing the strainer element. The free hole area of the screen element in the strainer body is from two to four times the area of the pipe connection, depending on the service and the size of strainer used. Makers claim strainers are suitable for steam pressures to 250 lb per sq in. and temperatures to 450F. For cold liquid lines the pressure rating is 400 lb per sq in.

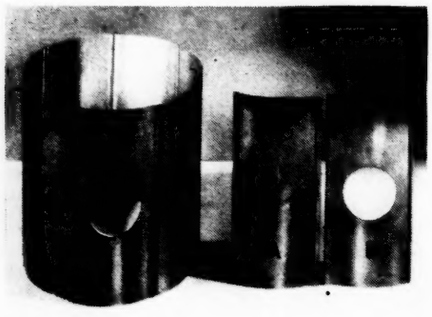
MADE BY—*Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Mich.* **470**

Combustion Chamber

NAME — Gregory stainless steel high temperature adjustable combustion chamber.

PURPOSE—Combustion chamber for use with round, square, or rectangular fireboxes.

FEATURES—Makers claim that because there are no dead spaces with this combustion chamber starting noise is eliminated. Unit, it is



claimed, instantly reflects heat to bring improved combustion. Mirror-like finish is said to shed carbon and soot. Two type panels are required to form any size chamber. The *A* panel is standard and *B* panel has either 4, 5 or 6-in. knockout for burner nozzle.

MADE BY — *J. G. Gregory Co., Burner Division, 228 No. LaSalle St., Chicago 1, Ill.* **471**

Convect-O-Base

NAME—Convect-O-Base radiation.

PURPOSE — A convector which can be installed after the piping is completed.

FEATURES—Device is a hollow aluminum baseboard covering 1-in. pipe on which are clamped aluminum heat reflectors. Front cover is vented, top and bottom, to allow passage of cold air which absorbs heat from the reflectors and passes out top of baseboard. Reflectors are clamped on the pipe after the piping is completed. Device can be adapted to practically any room. Reflectors are detachable and can be added or removed, according to individual requirements. Front cover is a slip fit. Its functional design harmonizes with any decorating scheme. Using a small amount of water, time lag in heat pickup is held to a minimum. In typical installation, a 1-in. pipe loop is run along outside walls four inches off floor. Loop can be continuous, running from one room to next, and can be trapped under doorways and entrances. No valves, fittings, bleeders, or special fittings are required other than the square-headed cocks which are used in multiple pipe loop systems to balance loops uniformly.

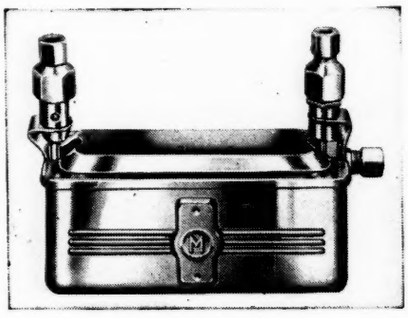
SIZES AND CAPACITIES—Capacity 3 square feet of equivalent steam radiator surface equipment per lineal foot of convector.

MADE BY—*Minimite Co., 623 S. Kildare Ave., Chicago 2, Ill.* **472**

Humidifier

NAME—Atomick humidifier.

PURPOSE—For adding humidity to indoor atmosphere conditions.



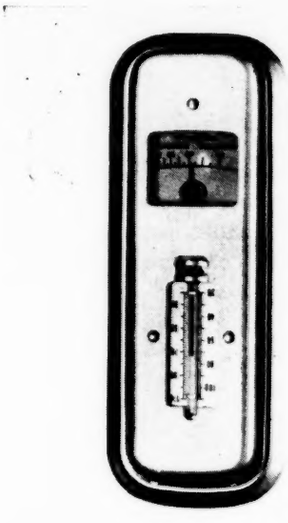
FEATURES — When connected for service with unit heaters, humidification starts when the unit heaters are in service and the humidifier continues to deliver water vapor directly into the warm air flow for as long as the unit heater is in operation. Humidifier may also be controlled separately by a humidistat and in hot dry climates can be used for cooling purposes. The single spray humidifier will atomize up to 1¼ gal of water per hour while the four-spray unit will handle four times as much. Humidifier is air operated and where a compressed air line is not available, a small air compressor may be employed.

MADE BY—*Maid-O'-Mist, Inc., 3218 N. Pulaski Rd., Chicago, Ill.* **473**

Humidity Indicator

NAME—Friez model 182 relative humidity and temperature indicator.

PURPOSE—Wall type instrument for



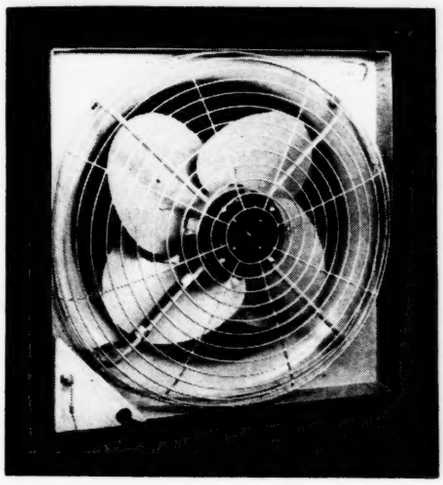
indicating relative humidity and temperature.

FEATURES — Relative humidity is shown directly in percent from 10% to 95% and the makers claim an accuracy of 5% or better. The humidity-responsive elements are made of specially treated human hair and are said to provide quick response to humidity changes. Air temperatures are indicated from 20 to 110F.

MADE BY—*Friez Instrument Div., Bendix Aviation Corp., Baltimore 4, Md.* **474**

Exhaust Fan

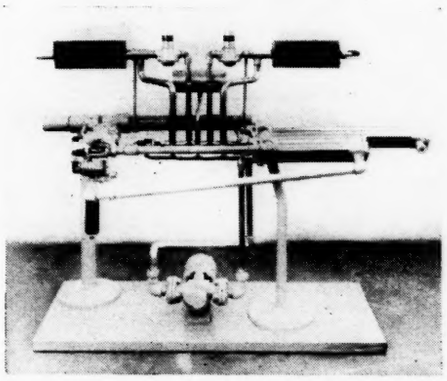
NAME—Fresh-air-maker.
PURPOSE—Exhaust fan for use in homes, stores or factories.
FEATURES—Unit has a square, steel frame with wide reinforced flanges pierced for mounting. Four motor-support arms are welded to the motor saddle for firmly supporting the motor and fan assembly. Rubber mounting of the motor insulates motor from frame to eliminate transmittal of vibration. A three-speed motor is provided.
SIZES AND CAPACITIES—In 12, 16 or 20 in. sizes with respective air



ratings of 1250, 1600 and 2500 cfm.
MADE BY—*Schwitzer-Cummins Co., Indianapolis 7, Ind.* **475**

Car Heating System

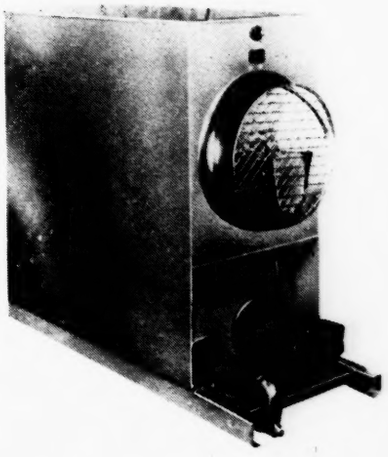
NAME — Vapor Car liquid heating system.
PURPOSE — Liquid heating system for railroad cars that provides both heating and cooling.
FEATURES — Heated liquid, a balanced solution of Prestone and water, is circulated through a heat exchanger, then through the cooling exchanger. From there it goes to the valves inside the car from an underneath loop with the loop so arranged that a single pump can take care of an entire car and various zones within the car. The cooling exchanger, when leaving a hot area, automatically goes into operation and supplies lowered tem-



peratures in whatever cars and/or zones that require a temperature change. Illustration shows the major parts of the system including a pair of heat exchangers, fin radiation, and a pump to circulate the liquid through the system.
MADE BY—*Vapor Car Heating Co., Chicago, Ill.* **476**

Cooling Tower

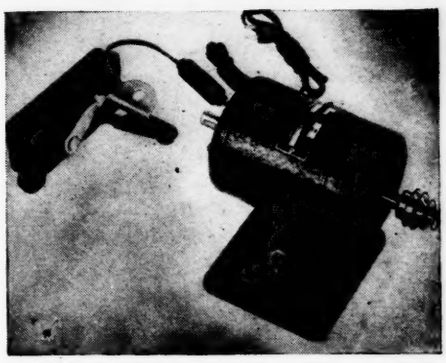
NAME—Aquatower.
PURPOSE—Compact, packaged cooling tower for use in air conditioning and refrigeration work.
FEATURES—Designed with standard components so that no special pumps or motors are required. Unit is available with or without pumps and all parts are readily accessible for servicing. A sturdy



fan guard insures safe operation. Unit has been engineered for indoor or outdoor service. It is sent out completely assembled for ready operation.
SIZES AND CAPACITIES—Three to 15 tons of refrigeration.
LITERATURE AVAILABLE — Bulletin AQ-47.
MADE BY — *The Marley Co., Inc., Kansas City 15, Kans.* **477**

Small Electric Motor

NAME—L & R 1/10 hp motor.
PURPOSE — For general use calling for an electric motor of this size.
FEATURES — Motor which operates on a-c or d-c is rated at 1/10 hp at 5000 rpm. A $\frac{3}{8}$-in. shaft extends in two directions and motor has double shielded ball bearings that do not require oiling. Outside diam-



eter of motor is $3\frac{1}{4}$ in. Motor is mounted on a pedestal, is supplied with three-step pulley, foot rheostat, and reversing switch.
MADE BY — *L & R Manufacturing Company, 577 Elm Street, Arlington, N. J.* **478**

Welding Rod

NAME—Airco No. 23A silicon-copper welding rod.
PURPOSE—For use in copper welding.
FEATURES—Makers claim that this rod will provide stronger welds than the deoxidized copper analysis previously offered. Rod is said to provide an excellent color match on copper.
SIZES AND CAPACITIES—Available in 1/4, 3/16 and 1/8-in. diameters.
MADE BY—*Air Reduction Sales Co., 60 East 42nd Street, New York 17, N. Y.* **479**

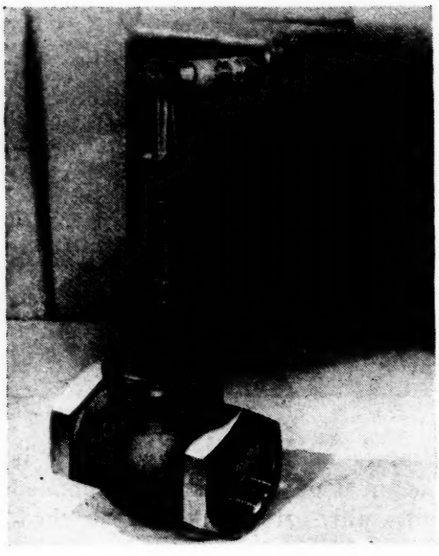
Motorized Valve

NAME—Motorized valve.

PURPOSE—For providing motorized operation of valves, burners and dampers.

FEATURES — Valve enables automatic, two-position operation of valves for steam, air, gas, oil or water. An induction type motor develops a positive power drive through a worm and spur gear reduction. Limit switches are adjustable. Valve requires a three-wire thermostat, pressure switch, or similar instrument for actuation.

SIZES AND CAPACITIES—Valves are supplied in globe-type, all bronze, either single or double seated in



sizes ½ to 2½ in., or butterfly valves with iron body and bronze trim in sizes 1 to 4 in.

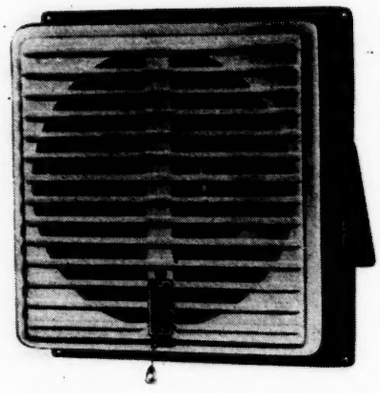
MADE BY—*Automatic Temperature Control Co., Inc., 34 E. Logan St., Philadelphia 44, Pa.* **480**

Exhaust Ventilator

NAME—Fasco ventilator.

PURPOSE — Exhaust ventilation for kitchens, laundries and similar rooms.

FEATURES—One control opens and closes the outside door which in turn starts and stops the fan automatically. While adaptable to many wall thicknesses the standard ventilator fits walls from 5 to 9½ in. thick, and can be used by means of



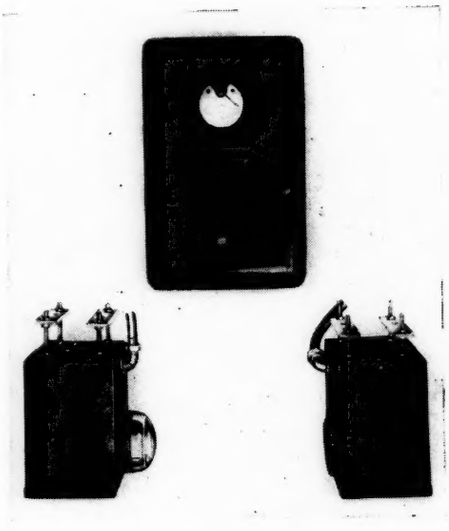
an extra sleeve for wall thicknesses up to 13⅞ in. Outside door, when closed, is held tight against a rubber gasket. Ventilator is powered by a shaded pole motor and extra oil reservoir. Voltage is 115 volts, 60 cycles a-c and can also be obtained for operation on 230 volts, 60 cycles a-c. The unit provides 405 cfm.

MADE BY—*F. A. Smith Manufacturing Co., Inc., 255 N. Union St., Rochester 7, N. Y.* **481**

Smoke Indicator

NAME—Ess smoke indicator model SGL-JS.

PURPOSE — Safeguard for proper combustion.



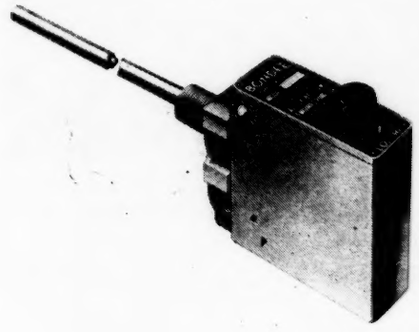
FEATURES — Indicator which operates on the principles of light entering through a stack, across a smoke passage to show how much light is cut off by the smoke. Exact condition of gases is read on the meter. At any predetermined amount of smoke, a light signal is displayed or a bell may sound as an audible warning. The meter which is used in connection with the photo-electric cell is marked with 100 points on either side of the center. By adjusting the sensitivity of the instrument and the lamp rheostat, this center point can be made to represent the most economical condition.

MADE BY — *Ess Instrument Co., Bergenfield, N. J.* **482**

Thermostat

NAME—Bondee thermostat.

PURPOSE—For use in processing work where a close differential is essential.

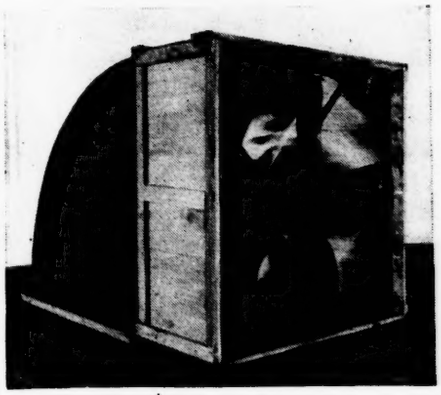


FEATURES — Unit has cast brass body, rod and tube thermal element, stainless steel pivot pin and a stainless steel spring. Spring provides for over or under travel when subjected to temperatures outside the calibrated range. It has an adjustable range of 30F to operate within any 30 degrees specified by the customer, from sub-zero to 300F. Electrical contacts are snap acting and have capacity of 10 amperes, 125 volts, or 5 amperes, 250 volts, a-c. It can be supplied with either single or double throw switch. Thermostat is provided with thermal element 6, 10, 16, 24, or 30 in. long.

MADE BY—*Alloy Bellows Engineering Co., 935 Greyton Rd., Cleveland 12, Ohio.* **483**

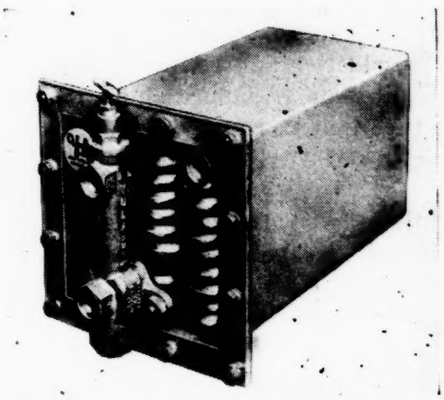
Suction Box

NAME—Healthaire suction chamber.
PURPOSE—To cut down the cost of attic fan installation by providing a suitable suction chamber.
FEATURES—All necessary hardware and parts are supplied so that the chamber can be easily assembled. Reinforced aluminum metal construction is used. The curved back provides an easy air flow and tends to reduce noise.
MADE BY—*Johnson Fan & Blower Corp., 1319 W. Lake St., Chicago 7, Ill.* 484



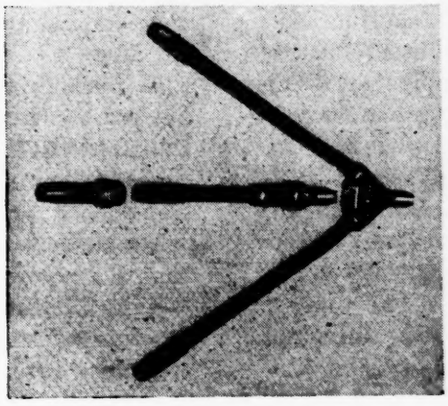
Mixing Valve

NAME—Yula combination mixing valve.
PURPOSE—Combination mixing valve in head heater for use without a storage tank, to provide a constant supply of hot water.
FEATURES — Domestic cold water enters the coil and travels through seamless drawn copper tubes. The boiler water enters the shell and flows between and around the tubes. The valve is a thermostatic three-way unit that operates on the principle of liquid expansion. The thermostat contains a special non-corrosive mineral oil which has a large volume increase for each degree of temperature rise. The bellows is protected by a copper tube casing to prevent kinking or jamming.
MADE BY — *Yula Water Heaters, Inc., 166 W. 225th St., New York, N. Y.* 485



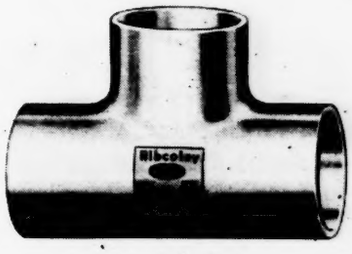
Brazing Tips

NAME—Multiflame brazing tips.
PURPOSE—Tips for use in silver brazing, high temperature brazing, and in the application of hard-facing alloys.
FEATURES—Tips are said to operate satisfactorily with oxyacetylene, oxyhydrogen, or natural gas. Tips are said to provide even heating throughout a joint area and to increase the speed of brazing operations when compared with a standard single flame tip.
MADE BY—*Air Reduction Sales Co., 60 East 42nd Street, New York 17, N. Y.* 486



Pipe Fittings

NAME—Nibcoloy.
PURPOSE — Fittings for use with light gauge tubes.
FEATURES — Fittings are available in inconel, nickel, monel, and stainless steel 304, 347 and 316. A lock ring presents a method of joining with silver brazing of soft solder.



This ring supports the joint during the joining operation and reinforces it afterwards. Line of fittings includes tees, elbows, reducers, adapters, and so on.
SIZES AND CAPACITIES — Available from ¼ in. to 4 in. O.D.
MADE BY—*Northern Indiana Brass Co., Elkhart, Ind.* 487

JULY, 1947

TO OBTAIN FURTHER INFORMATION ON NEW EQUIPMENT

On the list below, circle the item number of the equipment in which you are interested, using the number found at the end of each item.

467	468	469	470	471	472	473
474	475	476	477	478	479	480
481	482	483	484	485	486	487

Print your name and address, detach and mail to

Editor, HEATING AND VENTILATING, 148 Lafayette St., New York 13, N. Y.

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NEWS OF THE MONTH

BUILDING SUPPLY SHORTAGES

are not the result of skulduggery, according to one manufacturer's analysis of Commerce Department figures.

Shortages are not indications that somebody is hoarding or building up stocks or that manufacturers are holding down production waiting for prices to get higher, according to a Crane Co. analysis of building materials production figures. Manufacturers are shipping material out as fast as they can. Many wholesalers are loading fixtures directly on trucks for delivery, before they get a chance to bring the items inside their own doors. The truth of the matter is that demand exceeded supply several years back, and supply has never since been able to catch up.

The plumbing industry is manufacturing most items at an unprecedented rate, as U. S. Department of Commerce figures show. According to this agency, production rates of such items as soil-pipe, bathtubs, sinks, lavatories and water-closets had reached all-time highs by December of 1946. But the shortages that still exist are real and undeniable. A look at some actual figures will reveal why the shortages are there in the face of industry's tremendous production.

• **SOIL-PIPE.**—Let us look at the soil-pipe situation. In January, 1946, soil-pipe manufacturers had on hand orders for 272,473 tons of soil-pipe and fittings. This backlog kept getting higher until it reached a peak of 356,636 tons in October. Since then the manufacturers have been steadily chopping away at these orders, reducing the unfilled order totals by 2,900 tons in November, 4,100 tons in December, 6,400 tons in January, and 17,000 tons in February! In March the reduction in unfilled orders dropped off to 6,700 tons. But January and March soil-pipe shipment figures showed greater production of this material than at any other time in history—52,339 tons in January, 51,827 tons in March; February, a short month, was slightly lower.

• **PLUMBING ITEMS.**—The same is true of other plumbing items. Production is at an all-time high, yet we are soaking it up day by day, just as fast as we can make it. The Department of Commerce has given us index figures by which present production can be compared roughly with prewar production. The production index for all items in 1939, taken as the basic and normal year, is 100. Increases in production can be seen quickly by looking at the latest available index figures, those for February, 1947: for bathtubs, 158.1; for sinks, 221.1; for lavatories, 172.3; for water-closets, 137.1. How encouraging that looks! And we *can* be encouraged by these figures to a certain extent, too, because now we find that many scattered items *are* in plentiful supply—for instance, lavatories, steel enameled baths, brass fixture trim and brass valves. Sinks are in fair supply. But still hard to get are water-closet combinations, enameled-*iron* baths, hospital fixtures, steel cabinets for sinks, boilers and radiation, steel pipe and malleable fittings. It's that old glutton "demand" creeping in again, says Crane.

The fact that the supply situation on sinks has eased somewhat is explained by production figures here also. Estimated requirements for 1947 are 2,101,000. The industry produced 913,000 in the first quarter. At this rate we have a potential of 3,752,000 sinks for this year. When the back-order file has been cleaned out, it appears that by year's end nobody will hunger for sinks.

• **RADIATION.** — Radiation, cast-iron or convection, and plumbing pipe and fittings remain the hardest nuts to crack. Shortages of pig iron have made it almost impossible to bring production of radiation equipment up to normal. This is evidenced by the index figure of 84 at the end of February, 1947. The 1941 index for cast-iron radiation was all the way up to 140. With production slowed down to this extent, we can see how impossible it is for anyone to accumulate an inventory.

According to the American Iron and Steel Institute, last year over 4¼ million tons of butt-weld, lap-weld, electric-weld and seamless pipe and tube were produced for direct use. In the first two months of 1947, 775,260 tons were produced for direct use (for use as pipe, not for assembly into other manufactured items). At this rate 1947 production could surpass 1946 by considerably more than one-half million tons. Sounds like tremendous production—and it is—but requirements are even more tremendous. American Iron and Steel Institute records show that every major manufacturer of pipe has his order books full all the way to 1950. In other words, even if no additional orders came in from now on, it would take until 1950 to supply all of the pipe *already* ordered.

• **CATCHING UP.** — We could cover numerous other items, says Crane, but the story would be the same. Production is high, but things don't seem to be available. When we analyze production and consumption figures, we find that they are running very close, with production behind consumer needs in many instances. But a year ago consumer needs were running far ahead of production on most counts. A limited amount of finished product is being exported. Certain types or designs are more in demand than others, so that, although the overall production figures for a particular class of fixture or equipment are approaching the encouraging side, the type that you are trying to obtain may not be available in quantity—sometimes may not be available at all.

At present the outlook is good. Order backlogs are being whittled down a little more each month. It is hoped that by the end of 1947 most items will be in full (although still not plentiful) supply, so that contractors will be able to order "out of stock" instead of from "next month's production."

DR. CARRIER

tells British engineers of twenty major advances in the last decade of air conditioning.

Dr. Willis H. Carrier, Chairman of Carrier Corp., and a pioneer in the air conditioning industry, told the centenary celebration of the British Institution of Mechanical Engineers on June 10 that lower initial costs, increased refrigeration safety through the use of new refrigerants, and a general acceptance of huge centrifugal refrigeration machines for large-scale air conditioning projects were outstanding among 20 major advances in this field during the last decade.

• **OUTSTANDING.**—Dr. Carrier divided his list of 20 "outstanding developments" in air conditioning and refrigeration into 13 in the field of equipment, five in application and two in theory, as follows:

Air Conditioning and Refrigeration Equipment

1. A new type of air filter, combining filtration with electrostatic precipitation of dust.
2. A practical absorption filter, suitable for use in ventilation.
3. Wider adaptation of new refrigerants to air conditioning and allied processes.
4. A rapid trend in refrigeration installations toward replacement of large reciprocating machines by centrifugal compression units.
5. Adaptation of the steam jet refrigerating system to railway coaches.
6. "Lo-Fin" Tubing for condensers and evaporators.
7. A new development of centrifugal compressors for refrigeration for low temperatures.
8. A direct expansion heat transfer unit for cooling and dehumidifying air at extremely low temperatures.
9. Increased use of cooling coils as a substitute for the spray-type air conditioner for dehumidification in air conditioning installations.
10. Development in refrigeration for air conditioning of multi-cylinder high-speed reciprocating compressors with variable capacity control.
11. Portable self-contained low temperature air cooling units for cold storage air conditioning and refrigeration.
12. Improvement in design and in consequent performance of axial flow fans, leading to their greater application for ventilation purposes.
13. Radiant heating and cooling.

Air Conditioning and Refrigeration Applications

1. Application of air conditioning with refrigeration to windowless factories processing non-hygroscopic materials.
2. Addition of refrigeration to the air conditioning of cotton mills.
3. Dehumidification and humidity control for blast furnaces.
4. Improved systems for air conditioning multi-room buildings and passenger ship staterooms.
5. Dehumidifying and cooling the air in a barometric wind tunnel for airplane engine testing.

Theory in Air Conditioning Applications

1. Additional data on the relation of radiation, convection and evaporation in heat disposal from the human body.
2. Development of the theory of the air jet as applied to air outlets used in ventilation.

It would be folly to conclude that all noteworthy advances in this art have now been made, said Dr. Carrier. In general, however, no revolutionary changes may be expected except possibly in the field of refrigeration or water-cooling for air conditioning purposes. The value of air conditioning both in industry and for human comfort will be increasingly recognized.

PIPING CONTRACTORS

hold four-day convention at Cincinnati with sessions on radiant heating and welding.

Predictions for a good business year in 1948 were made by *Dr. N. R. Whitney,* economist, The Proctor and Gamble Co., in a talk at the opening session of the fifty-eighth annual convention of the Heating, Piping and Air Conditioning Contractors National Association held June 11-14, Hotel Netherland Plaza, Cincinnati, Ohio. Registration totaled 510 members and guests.

• **BOOM AHEAD.**—Dr. Whitney said that business reports for the first quarter of 1947 pointed to a boom period with 57,000,000 persons employed, wages highest and work-week shortest in history. It is difficult to get wokers now and there is a very acute steel shortage. Low interest rates plus taxes are directing private capital into government securities so that industry is having a difficult time to finance projects. At the end of the first quarter this year there was the start of a downward trend with wholesale prices starting to slip. He predicted a downward trend not exceeding 30% for the balance of the year, and after this mild recession a new high level of prosperity in 1948. During the next ten years he predicted an increase in the quantity of goods and also an increase in the variety of occupations.

A talk on the steel pipe situation was made by *Henry J. Wallace,* general manager of sales, National Tube Co. In a very frank discussion, he presented a picture of existing conditions and told of what is being done to curtail the development of black market conditions in steel pipe.

• **PANEL HEATING.**—A panel discussion on home heating and panel heating was held on the afternoon of the second day. *Peter B. Gordon,* chairman of the society's Radiant Panel Heating Committee served as chairman of this discussion. A talk on Temperature Controls in Panel Heating Installations was given by *John Haines,* vice-president, Minneapolis-Honeywell Regulator Co., Minneapolis, Minn. Experiences with

Panel Heating from the Contractor's Viewpoint was presented by *George P. Nachman,* Cleveland contractor. Following these three talks, there were a number of questions directed from the floor to the "board of experts."

Frank E. Kise, vice-president, Williamson Heater Co., spoke on Interesting Junior Employees in Management.

• **WELDING.**—The major part of the Friday afternoon session was devoted to welding. Beside the report of Committee on Welding by Chairman *Richard S. Tobin,* a Report of National Certified Pipe Welding Bureau by Chairman *William W. Murray, Jr.,* and a report on the Standard Manual on Pipe Welding by Chairman *John W. Zink,* there was a paper on Welding Stainless Steel Piping by *J. D. Mattimore,* chief engineer, Product Engineering and Research, Tube Turns, Inc.

Beside the papers and features mentioned, there were reports of the various standing committees. The convention closed with the installation of officers on Saturday morning.

Entertainment features included a midnight buffet for those who arrived on Tuesday evening, an informal dinner and get-together party on Wednesday evening, dinner-dance on Thursday, and a river cruise to Coney Island which also included a dinner and dance.

• **NEW OFFICERS.**—The new officers of the association are: *James M. Anderson,* Philadelphia, Pa., president; *Thomas L. Eagan,* Washington, D. C., vice-president; *Louis F. Hudepohl,* Cincinnati, Ohio, treasurer; *Joseph C. Fitts,* New York, N. Y., secretary.

INVENTIONS WANTED

by Army, Navy and Coast Guard are listed by National Inventors Council.

The National Inventors Council has listed twenty-five inventions wanted by the Army, Navy, and Coast Guard. Some pertinent items are these:

• **LOW HORSEPOWER GAS TURBINE.**—The requirements are for a lightweight, variable-speed gas turbine of approximately 0.8 lb per brake horsepower with a fuel consumption of less than one pound per brake horsepower hour and capable of utilizing low-grade fuels. Various sizes up to 200 hp are required. Immediate applications include use for operating water and petroleum pumps and electric power generators.

• **FUELS AND LUBRICANTS.**—The use of engineering equipment in areas of extreme temperatures presents new problems in the starting and operating of gasoline and Diesel engines. New types of fuels and lubricants which will allow normal starting and dependable operation of engines are the outstanding requirements. The use of additives for standard fuels and lubricants also requires study and development.

• **COMBINATION PLASTIC AND ALLOY.**—The use of plastics or combinations of metal alloys and plastics in the fabrication of bridge structural members, pontoons, and pneumatic floats would improve the speed and ease of erecting bridges. The decrease in weight of the unit would require less personnel for handling and increase the carrying capacity by decreasing dead weight. Suitable plastic landing mats to replace mats made of critical and more expensive metals are desired.

• **LIGHT STRUCTURAL METALS.**—Light-weight, high-strength, non-corrosive structural metal alloys with low flammability characteristics are wanted as substitutes for all engineering equipment now made of heavier metals, such as engine-generator sets, air compressors, pontoons, boats, and bridge structural members.

• **PREVENTION OF CORROSION.**—The solution of the problem of fungi and corrosion prevention on all types of material, particularly electrical equipment, is of vital importance.

• **COLD-RESISTING ALLOY.** — At temperatures below 25F steel and aluminum fracture easily. An alloy or treatment is wanted to counteract this change occurring at low temperature.

• **STANDBY HEATERS.**—Under Arctic temperatures of —65F, heaters of both the hot air and engine coolant type are a critical need.

• **SHIP DRAFT GAUGE.** — The Transportation Corps wants a draft gauge to provide accurate readings while the vessel is at sea.

• **OIL BURNER IGNITER.**—There is need for a cleaner, safer, more efficient and positive means than the use of present standard hand oil torch to light fuel burners on ship's boilers.

• **FUME DETECTOR.**—There is an urgent need for a combustible gas alarm as a warning device against bilge explosions on gasoline-engine-propelled boats.

• **LIGHT DIESEL.**—The Coast Guard wants a light-weight Diesel engine of eight to ten brake horsepower at 3,000 rpm, weighing not over 5 lb per brake horsepower to take the place of the gasoline engine of portable emergency fire pumps. Fuel consumption should be not more than five pounds per brake horsepower hour.

• **SCALE-FREE EVAPORATOR.**—For ship service distillation of boiler and drinking water. At present the output of evaporators falls off with the deposition of salt, scale and corrosion of metal .

STOKER MAKERS

stress competitive market at their 30th Annual Meeting at Lookout Mountain, Tenn.

The merchandising of automatic coal stoker equipment, new applications of electronic controls, the competitive fuel situation, the supply of coal for stoker application, and public relations were among the many subjects covered in the 30th annual meeting of the Stoker Manufacturers Association held at Lookout Mountain, Tennessee, June 16, 17, and 18.

• **OPTIMISTIC.**—An optimistic and aggressive attitude were reflected in the reports and discussions. *Walter Sormane,* retiring president of the association, declared "that automatic heat with coal is and will remain

the basic, most comfortable and economical type of heat for the great masses of American people."

B. R. Gebhart, chairman of the marketing committee of the National Coal Association, told the stoker manufacturers of the nation that the situation today "indicates a most encouraging and cooperative belief in the future of the automatic firing of coal . . . the coal producing industry is doing everything within its power to keep the gap bridged between supply and demand for premium grades of stoker coal. Given reasonable operating time and the expected increase in selective application of stoker coals, I am confident that our industry during this coal year can satisfactorily serve its stoker coal customers, both present and prospective."

• **MEMBERSHIP.**—*Marc G. Bluth*, executive secretary of the association for the past fifteen years, in reviewing recent nation-wide developments in fuels and fuel research stated: "There are on the horizon already signs of the greatest revolutionary developments in fuel and machinery in the history of the world and these major developments have exerted their influence in many ways and in many directions." Referring to the continued growth over the period of the past fifteen years in the membership of the association, Mr. Bluth stated that the membership is strong, and 135 delegates, representing stoker manufacturing firms, allied supplier firms, and allied industries, attended the meeting. *Claude A. Potts* served as general chairman of the meeting and was assisted by *Frank Hoke* on program and *Charles Cochran* on entertainment. The convention ended with the annual dinner, with *C. T. Burg* of Cleveland acting as toastmaster. Certificates of appreciation with appropriate gifts were awarded to retiring president *Walter Sormane* and retiring chairman of the engineering and research committee, *A. O. Dady*.

At a meeting of the newly elected members of the board it was decided to hold a membership conference late next October or early November, probably in Chicago.

AIR CONDITIONED FACTORY

and controlled process temperatures speed candy manufacture and foil summer shutdown.

In the newest candy factories, push-button temperatures are increasing candy output, bettering plant sanitation, and providing a higher quality and more uniform product, according to reports to the Refrigeration Equipment Manufacturers Association.

A typical example of how air conditioning and refrigeration are becoming increasingly important in the candy industry is the $1 million plant just completed by the Walter H. Johnson Candy Co., Chicago, which will be placed in operation this summer. The factory will be used for increased production of candy bars and other types of confections.

• **ZONED.** — Revealing several new departures in candy manufacture, the plant is air conditioned throughout, from executive offices to candy wrapping rooms. Containing 100,000 sq ft of space, the factory is divided into four separate sections — each with temperatures and humidity conditions regulated in accordance with the operations performed within the section. Conditions for different sections range from 60F with 45% R.H. to 80F with 50% R.H.

• **FOR INSTANCE.**—A temperature of 60F with 45% R.H. is maintained in the section where Powerhouse candy bars are made. Direct expansion coils are used in this section to maintain proper temperatures within the enrobing tunnel, where candy with chocolate coated centers is cooled. Water from city mains is chilled and and is used throughout the rest of the plant.

Refrigeration is also used to chill the water used in the candy "slabs", the large metal tables upon which candy is allowed to cool following its preparation in the kitchens. Chilled water at 35F is used to maintain a table surface of 37F.

DISTRICT HEATING MEN

talk things over in Atlantic City as NDHA holds 38th Annual Meeting.

The National District Heating Association held its 38th Annual Meeting at the Chalfonte-Haddon Hall Hotel in Atlantic City, June 24-27. Secretary-Treasurer *John F. Collins, Jr.*, reported over 100 members registered early the first day and predicted heavy attendance.

• **PROBLEMS.**—Chief points of discussion, as evidenced by the program, were problems relating to distribution, commercial relations, accessories, rates, station operation, and engineering. President *John F. Malone* greeted members at the opening session and pointed up the function of the association as a medium for exchange of ideas—increasingly important amid the growing legal and financial complexity of today.

• **PAPERS.**—The program was so organized as to include technical papers and discussion periods under the auspices of each committee reporting on the various aspects of district heating problems. For instance, the report of the commercial relations committee, *A. T. Veness*, Rochester Gas and Electric Co., chairman, included a paper on rights of way for main extensions, by Mr. Veness, and a paper, "Types of Temperature Control" by *O. J. Prentice*, C. A. Dunham Co. The latter paper was read in Mr. Prentice's absence by *Albert F. Metzger*, Allegheny County Steam Heating Co. Similarly, the report of the rates and regulations committee, *Harry A. Weitzman*, Rochester Gas and Electric Co., chairman, included a three-part discussion by Mr. Weitzman covering rates, accounting, and legal decisions. Other committee reports included papers on steam for air conditioning, maintenance costs, metering, main construction, corrosion, etc.

• **OFFICERS.**—At the final session on Friday, June 27,

the following association officers, elected at the opening session on Tuesday, were inducted to serve for the year 1947-48:

President, *Robert D. Martin,* New York Steam Corporation, New York, N. Y.; first vice-president, *Henry L. Martin,* Boston Edison Company, Boston, Mass.; second vice-president, *David W. Loucks,* Allegheny County Steam Heating Company, Pittsburgh, Pa.; third vice-president, *George H. Tuttle,* The Detroit Edison Company, Detroit, Mich.

Directors for a term of two years: *Thomas P. Brown, Jr.,* New York Steam Corporation, New York, N. Y.; *Percy A. Hyde,* Ohio Edison Company, Akron, Springfield and Youngstown, O.; *Milton A. Hanna,* The Ric-wiL Company, Cleveland, Ohio.

WALL CONSTRUCTION UNIT

featuring metal-clad insulation has first public showing in New York.

Introduction to the construction industry of a new low-cost building system was made June 5 and 6 at a special showing held at the Commodore Hotel, New York. Architects, engineers, builders and others viewed a completely assembled bay made up of units of the new system now in the final stages of experimental research.

• **BASIC UNIT.**—Key unit in the system is a metal-clad insulated wall section. A typical unit of this new wall, 52 in. square, weighs less than 200 lb. The same area of wall in conventional brick, or brick and tile construction would weigh from seven to twelve times as much. Although these new units are only 3½ in. thick, they are said to be almost six times as resistant to heat transmission as a solid masonry wall 13 in. thick.

Two of these 52-in. square units span from the ceiling of one floor to the ceiling of the next, so that on each floor the wall is divided horizontally into two parts. The lower half is a line of solid units bolted to the conventional reinforced concrete or steel frame of the building. The upper half of the wall may be windows or solid wall in any proportion that the needs of a particular building may dictate.

• **CO-OPERATIVELY DEVELOPED.**—The exhibit was sponsored by the New York Housing Trust, a privately-endowed, non-profit organization, which enabled architect William Lescaze and designer Robert Davison to undertake the necessary research.

R. A. Biggs and W. W. Krauss, development engineers for Pittsburgh Corning and Republic Steel respectively supervised engineering and fabricating of several prototype units of the new thin wall, using stainless steel and Foamglas, and including a complete ceiling-to-ceiling section of a multi-story building.

Commenting on the unit both Mr. Biggs and Mr. Krauss emphasized that there was still work to be done before it could be universally adaptable. In their opinion it is an example of what can be accomplished in modern construction using modern materials and methods.

Other prototype units of slightly different design were also on display. They make use of copper, aluminum and stainless steel for the outside surfaces, combined with Foamglas, microporite and vermiculite as insulating cores.

MOW-CURED BALES

of hay are produced with power-driven fan. Heated air blown through mow in some USDA experiments.

Mow curing of baled hay is one of the variations in the many recent experiments using power-driven fans to cure hay. The object of all these experiments is to prevent weather damage to hay, whether from sun or rain and dew, says the U. S. Department of Agriculture.

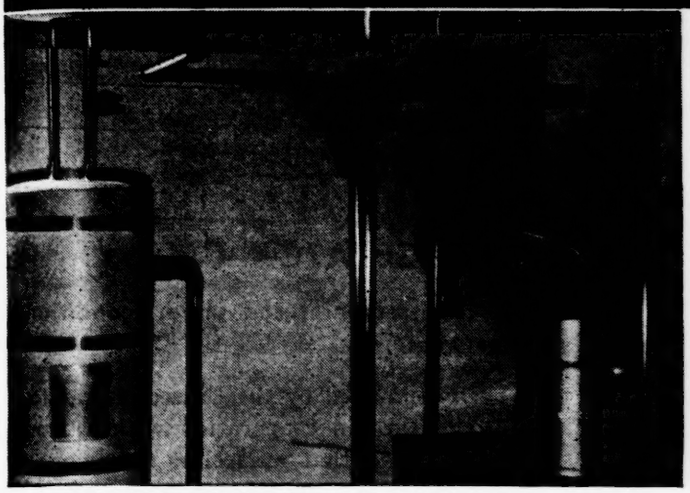
• **PILING.**—In experiments in which the Department has cooperated with the Iowa Agricultural Experiment Station the last two seasons, excellent results followed careful piling of baled hay over an air duct system on the mow floor through which air was blown. When bales of hay were piled on their edges with 2-in. air spaces between the bales and with alternate layers at right angles, alfalfa hay of 25 to 30% moisture dried satisfactorily with little loss of color and with the saving of most of the valuable leaves. In one test, bales dumped helter skelter, without care in piling, cured satisfactorily except for a few bales having high moisture content, which developed musty spots.

This hay was from first and second cuttings of alfalfa baled in the field before the sun had bleached the hay and while it was moist enough (25 to 30%) so that the baler could pick up the hay without losing many leaves. At this stage the hay is almost three-fourths cured.

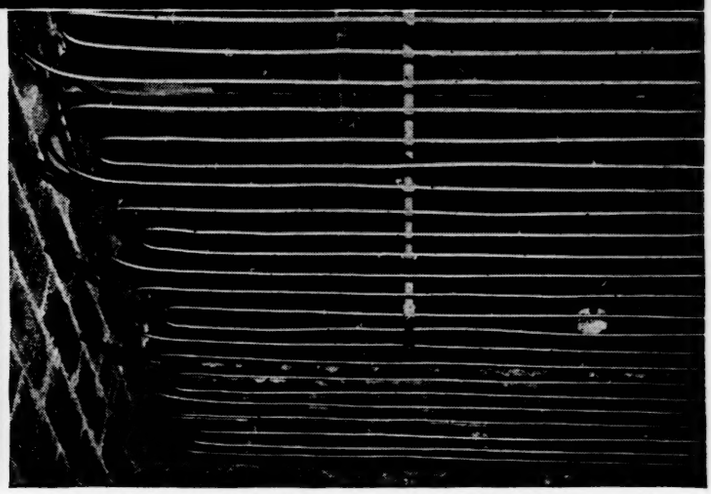
Evaporation of the moisture gave a cooling effect, and recording thermometers inside the bales registered temperatures several degrees cooler than the fan-driven air. In the Iowa experiment hay from the first crop of alfalfa was well cured, and the bales were moved elsewhere before the second crop was cut.

• **TYPES.**—Promising experiments with the drying of full-length hay, and others with chopped hay, some with heated air and some with air at normal temperatures have also been conducted at several experiment stations. The place of this method of making hay in American agriculture has, however, not yet been fully determined, says USDA agricultural engineers. Many types of installations are being tested and it is expected that some of them will be adopted rather widely by farmers in some regions, farmers selecting the equipment that will best suit individual needs and regional climates. In humid regions farmers may

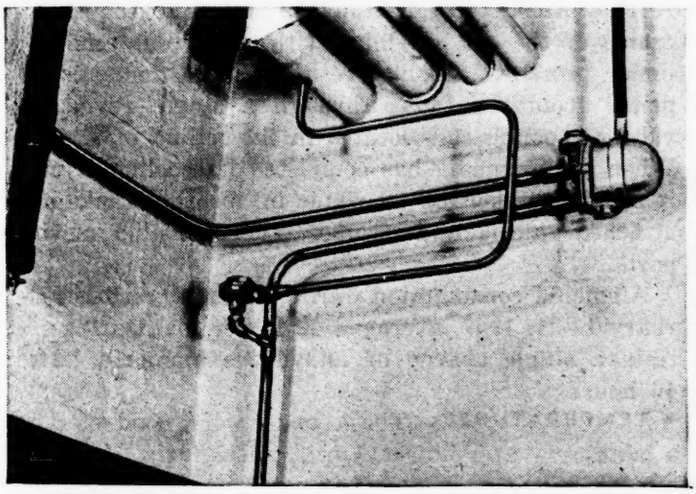
REVERE COPPER Makes Any System a Better System



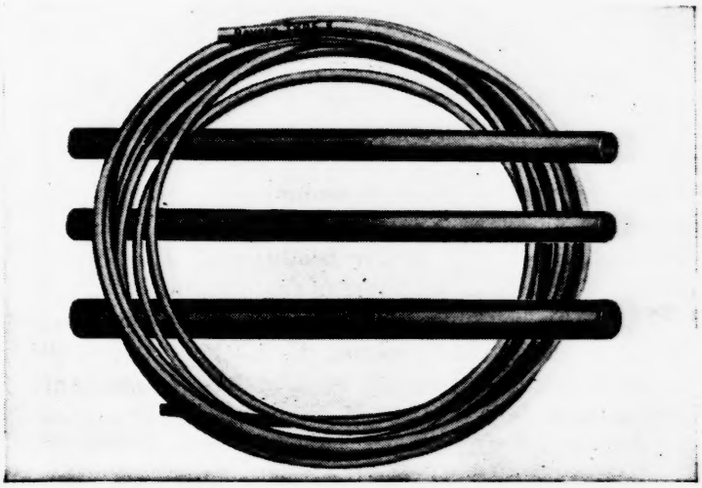
HOT WATER HEATING. Revere Copper Water Tube has a permanently smooth interior finish that insures unrestricted flow of water through the lines in either forced circulation or gravity systems. Joints made with solder or compression fittings help further to cut down friction loss, provide economy in cost of installation and give a much neater appearance of the finished work.



RADIANT PANEL HEATING. The smaller outside diameters and light weight of Revere Copper Water Tube are a distinct advantage in wall and ceiling panels. The use of long lengths of soft, easy-to-bend Revere Copper Water Tube provides ease of fabrication right on the job. And you can be sure, with Revere Copper Water Tube, of years of maintenance-free service.



STEAM HEATING. Steam return lines of Revere Copper Water Tube stay corrosion-resistant. Branches to radiators and convectors can be bent to produce smooth flow, which saves both the cost of fittings and the time of connecting them. Where joints are necessary, they are quickly made—even in cramped spaces—with solder or compression fittings.



THE REVERE NAME and the type, which are stamped on Revere Copper Water Tube at regular intervals, are your assurance of full wall thickness and the close gauge tolerances so essential for tight sweated joints.

You can specify or install such other long lived Revere materials as Red-Brass Pipe; Sheet Copper and Herculoy for tanks, ducts, pans and trays; Dryseal Copper Refrigeration Tube (dehydrated and sealed); Copper oil burner, heat control and capillary tubes. The Revere Technical Advisory Service, Architectural, is always ready to serve you. Call your Revere Distributor.

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Founded by Paul Revere in 1801

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Mills: Baltimore, Md.; Chicago, Ill.; Detroit, Mich.; New Bedford, Mass.; Rome, N. Y.—Sales Offices in Principal Cities, Distributors Everywhere.

TROUBLE Always Costs More Than REVERE COPPER WATER TUBE

make plans to dry nearly all their hay or put it up as grass silage, either method probably being better than field curing. In regions where weather is more favorable, the good farmer may have only a part of a loft equipped to dry hay that can be rushed in ahead of threatening rain.

NEWS BRIEFS

• A temperature difference of only one degree between walls and room air will cause dust and dirt to settle on the walls, Westinghouse scientists have discovered. Nail heads and other local cold spots attract dirt readily.

• FHA has approved mortgage insurance financing totaling $275 million for construction of 3,010 rental homes by American Community Builders, Inc., on their 2400-acre Matteson, Ill., site (H & V: 3:47: p. 108) 30 miles from Chicago's Loop.

• Development of an automatic hot water heater which derives its heat from surrounding air by means of a Freon refrigeration cycle is announced by Hotpoint, Inc. This heat pump is still in the experimental stage and may not be ready for general distribution for some years. Several utilities will cooperate in home installation testing of the device.

• The signing of the largest single contract for complete air conditioning in the history of the textile industry ($500,000) covering the three U. S. mills of Gotham Hosiery Co., Inc., has been announced jointly by Gotham and Carrier Corp. Installation, in both office and manufacturing space, will be largely complted this summer.

• Two passenger steamships under construction at the Bethlehem Shipyard, San Francisco bay, for American President Lines will be air conditioned throughout. George Sharp, naval architect is the designer.

• Should the heat pump prove practical and popular for space heating and cooling, the mechanical refrigeration industry can supply 90% of basic component parts without delay says E. M. Flannery, newly elected president of REMA.

• Trade practice rules for the vertical turbine pump industry have been promulgated by the Federal Trade Commission and may be obtained on application to the commission.

• Henry St. Leger leaves his post in the Office of the United States Chief Counsel for the Axis trial at Nurnberg to accept appointment as first general secretary of the newly formed International Organization for Standardization (ISO) in which 23 countries are now participating.

• Ford, Bacon and Davis, Inc., of Chicago and New York, has been retained as architect-engineer-construction manager of the Argonne National Laboratory in DuPage County, Ill., with the University of Chicago as prime contractor with the Atomic Energy Commission for construction and operation of the laboratory in which 29 middle-western universities will cooperatively conduct nuclear research. Approximately 3700 acres have been designated for the site.

• The 1947-1948 president of The American Society of Mechanical Engineers will be E. G. Bailey, New York, vice-president of the Babcock and Wilcox Co., manufacturers of steam generating equipment. Mr. Bailey was nominated for the office at the semi-annual meeting of the society, held in Chicago, June 15-19. Nomination is tantamount to election, since only one nominee is named for each office. He will succeed Eugene W. O'Brien of Atlanta.

ICELESS REFRIGERATOR CAR

maintains low temperatures under conditions of summer heat.

Tests have been conducted by the United States Department of Agriculture in cooperation with various industry groups showing that an iceless refrigerator car can maintain temperatures of approximately 0F under conditions of summer heat, a temperature which cannot be reached under those conditions by the ordinary refrigerator car.

• **AMMONIA.**—The car, with 3-in. insulation on all sides, is equipped with a split-absorption system of refrigeration. Charged tanks attached to the underframe of the car are filled with anhydrous ammonia under pressure. Cooling results as the ammonia expands from the tanks through a single regulating valve to cooling coils located in the ceiling of the car. When the ammonia has completed its cooling circuit, it is finally taken up by water in absorber tanks also attached to the underframe. There are no moving parts.

Ammonia consumption during the test period, February 11-21, 1947, averaged 42 lb per hr. During the tests a single charge of anhydrous ammonia lasted 46 hours.

• **TEMPERATURES.**—These tests were conducted in a laboratory at Potomac Yards, Alexandria, Va., on a car loaded with cartons of frozen tangerine segments. Air temperature in the carhouse lab was maintained at 92F for the entire ten-day period. Temperatures of the air in the laboratory and the frozen tangerines inside the car were determined every two hours by means of thermocouples and electric resistance thermometers which allowed reading of temperatures without opening the car doors.

During the test period temperature of the frozen tangerines at the top of the load averaged from —4F to 1F. At the bottom of the load temperatures ranged from —3F to 3F. It was decided that improved wall and floor racks, allowing a freer circulation of air, will undoubtedly narrow the spread between temperatures at top and bottom of the load. When placed in the car the tangerines were all hard frozen, and were still hard frozen on removal at the end of the test period.

Simplifies Piping Installation With Arc Welding

By William P. Miller, Engineer Smith & Oby Co., Heating Contractors, Cleveland, Ohio

IN the piping systems of the new 7-story addition to the Cleveland Clinic, we are using arc welding for pipe joints and for making brackets and hanger frames to simplify installation.

Fig. 1 shows the welding of risers which run about 120 feet from the basement to the top floor. At left and center are 5-inch feed and return pipes for air conditioning chilled water. At right is a 4″ steam line. These horizontal welds in vertical pipe are made as shown in Fig. 2, using ⅛″ "Fleetweld 5" Electrode for first pass and 3/16″ for subsequent passes.

Fig. 2.

At each floor level the risers are held by the bracket frame shown in Fig. 3. It is made from 2″ x 2″ angle, corners formed by coping out one flange, bending and welding in one pass. Holding rings are cut from pipe and fillet welded to the frame. The frame is welded to the building structure as shown. Wedges hold the pipe until insulation is applied.

The basement support for these risers is shown in Fig. 4. The supports are made from the same diameter pipe, flame-cut to fit and welded in three

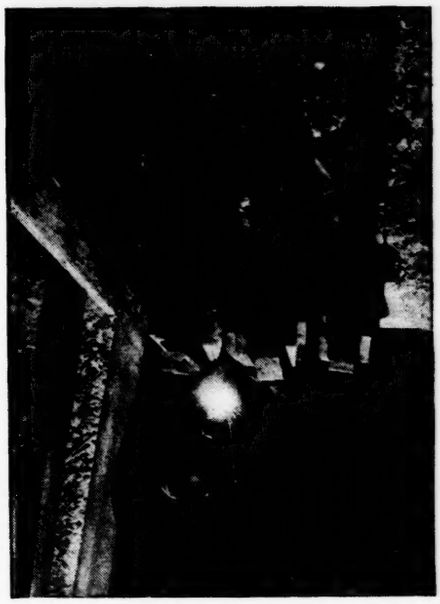


Fig. 1. Making finish pass on vertical pipe.

passes. A concrete column supports the three runs.

The frame—shown in Fig. 5—to hold copper water lines is made of ½″ x 2½″ bars, 28″ and 6¾″ in length. The ends are scarfed 45° and joined at corners with welds inside and outside. The plumbers' sleeves for different size pipes are cut from pipe and welded to the rectangle with continuous fillets. This frame fits over an opening in the concrete floor as shown.

We used "Fleetweld 5" electrode for all welding and a Lincoln "Shield-Arc

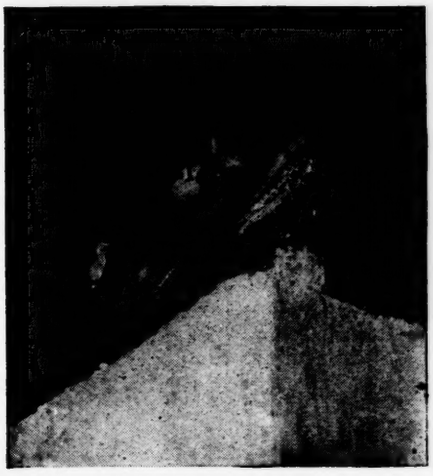


Fig. 4. Basement support for risers.

SAE" 300-amp. motor-driven welder, shown in Fig. 6.

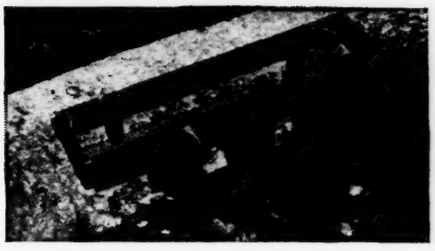


Fig. 5. Another frame for holding water lines at each floor level.

Specifications and procedures for "Fleetweld 5" and all other Lincoln electrodes are given in the Weldirectory. A free copy may be obtained by writing The Lincoln Electric Company, Dept. 482, Cleveland 1, Ohio.

Fig. 3. Bracket which holds risers in alignment is welded to I-beam ioist.

Fig. 6. Adjusting controls of "Shield-Arc" motor-driven welder. Running gear is handy on construction jobs.

Advertisement

Degree-Days for May, 1947

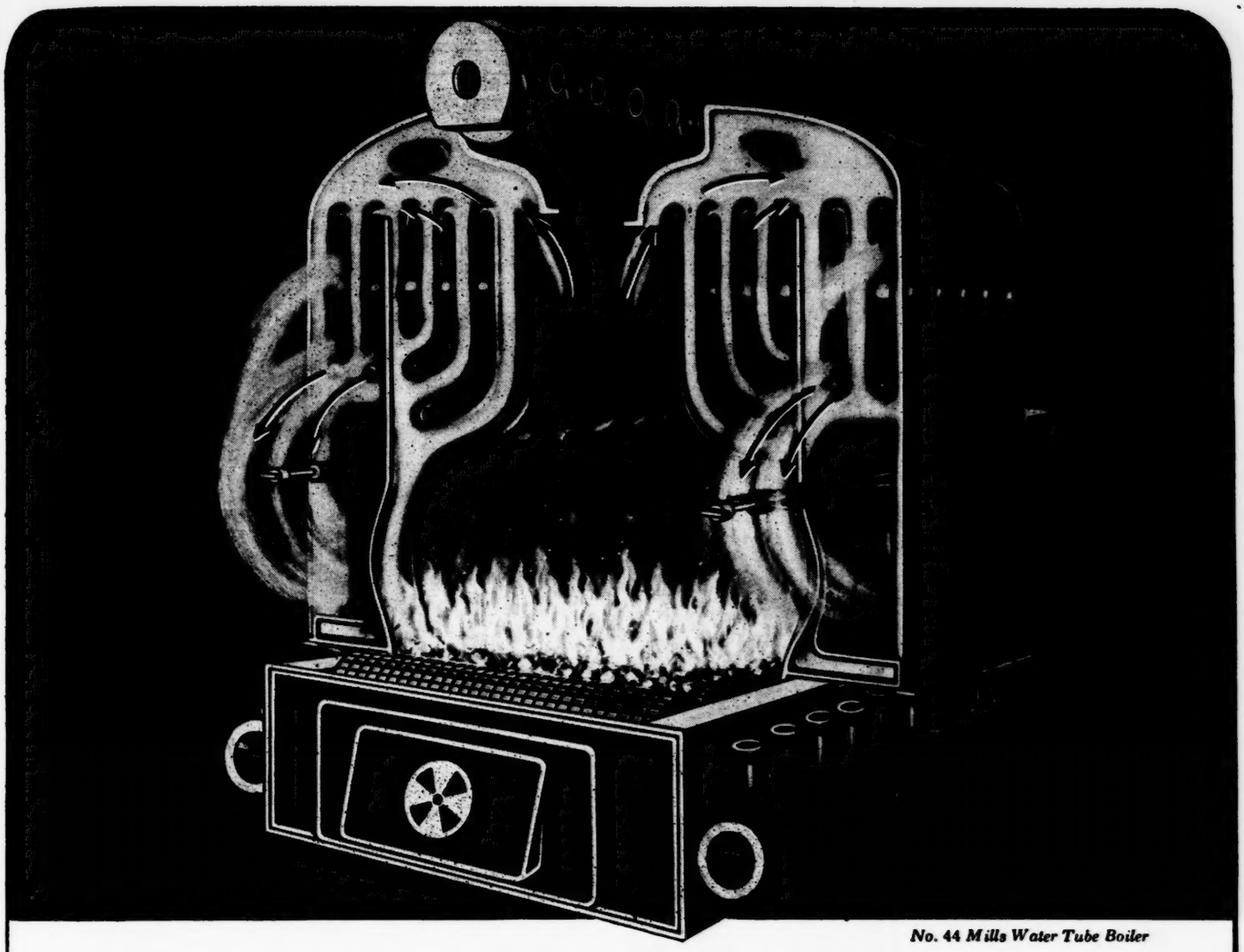
HEATING AND VENTILATING's 20th Year of Publication of Monthly Degree-Day Data

City	May			Cumulative, Sept. 1, 1946 to May 31, 1947			Heating Season[1]
	1947	1946	Normal	1946-47	1945-46	Normal	Normal
Abilene, Texas	23	27	0	2838	2388	2061	2061
Albany, New York	281	312	183	6629	6778	6580	6580
Albuquerque, New Mexico	48	76	40	4289	4242	4298	4298
Alpena, Michigan	574	509	465	7849	7635	8127	8299*
Anaconda, Montana	377	549	517	8057	7508	8010	8357**
Asheville, North Carolina	129	112	71	4075	3941	4232	4232
Atlanta, Georgia	43	22	0	2873	2778	2890	2890
Atlantic City, New Jersey	232	70	220	4519	4466	5176	5176
Augusta, Georgia	16	4	0	2191	2104	2161	2161
Baker, Oregon	230	364	431	6613	6726	6950	7163
Baltimore, Maryland	126	73	22	4071	4016	4533	4533
Billings, Montana	277	396	316	7311	6373	7059	7119
Binghamton, New York	260	277	254	6152	6310	6808	6808
Birmingham, Alabama	41	13	0	2783	2648	2352	2352
Bismarck, North Dakota	423	435	338	8929	8710	9147	9192
Block Is'and, Rhode Island	386	343	378	5269	5530	5689	5788
Boise, Idaho	146	230	236	5779	5626	5552	5552
Boston, Massachusetts	286	237	245	5409	5702	6045	6045
Bozeman, Montana	336	559	471	7705	7630	8195	8521**
Buffalo, New York	376	312	335	6451	6460	6810	6822
Burlington, Vermont	341	365	273	7370	7607	7511	7514
Butte, Montana	390	566	502	8474	8028	7926	8272
Cairo, Illinois	61	51	0	3783	3417	3909	3909
Canton, New York	407	387	334	7714	7894	8020	8020
Charles City, Iowa	364	327	224	7332	7165	7588	7588
Charleston, South Carolina	10	0	0	1724	1737	1769	1769
Charlotte, North Carolina	46	36	0	3028	2886	3120	3120
Chattanooga, Tennessee	72	24	0	3433	3175	3118	3118
Cheyenne, Wyoming	478	592	446	7467	6912	7340	7466
Chicago, Illinois	351	279	254	5992	5804	6018	6077
Cincinnati, Ohio	149	99	0	4467	4346	4684	4684
Cleveland, Ohio	283	247	220	5853	5603	6155	6155
Columbia, Missouri	143	144	22	4886	4464	4922	4922
Columbia, South Carolina	19	10	0	2296	2269	2364	2364
Columbus, Ohio	218	169	87	5220	5020	5398	5398
Concord, New Hampshire	349	351	298	7180	7435	7299	7353
Concordia, Kansas	194	211	65	5532	4746	5315	5315
Dal'as, Texas	11	18	0	2476	2103	2256	2256
Davenport, Iowa	257	222	118	6039	5881	6289	6289
Dayton, Ohio	248	214	81	5668	5462	5264	5264
Denver, Colorado	292	405	267	5861	5224	5874	5874
Des Moines, Iowa	(a)	224	118	(a)	5875	6384	6384
Detroit, Michigan	352	273	226	6280	6088	6490	6490
Devils Lake, North Dakota	508	452	424	9840	9869	9874	9970
Dodge City, Kansas	176	189	47	5272	4499	5035	5035
Dubuque, Iowa	316	240	149	6628	6393	6790	6790
Duluth, Minnesota	549	548	521	9290	9103	9267	9483
Eastport, Maine	520	525	543	7297	7708	7941	8520
Elkins, West Virginia	265	234	177	5718	5513	5697	5697
El Paso, Texas	1	5	0	2708	2538	2428	2428
Ely, Nevada	357	519	(a)	7357	7563	(a)	(a)
Erie, Pennsylvania	340	288	257	5845	5757	6273	6273
Escanaba, Michigan	602	504	490	8302	8026	8608	8771
Evansville, Indiana	157	122	0	4662	4284	4244	4244
Fort Smith, Arkansas	33	50	0	3336	3062	3147	3147
Fort Wayne, Indiana	285	275	130	6239	6120	5925	5925
Fort Worth, Texas	19	28	0	2532	2155	2148	2148
Fresno, California	4	16	0	2552	2435	2334	2334
Galveston, Texas	0	0	0	1407	1045	1016	1016
Grand Junction, Colorado	79	179	118	5654	5477	5548	5548
Grand Rapids, Michigan	381	278	174	6367	6211	6535	6535
Green Bay, Wisconsin	481	387	322	7744	7586	7825	7825
Greensboro, North Carolina	78	77	0	3758	3685	3529	3529
Greenville, South Carolina	39	37	0	3075	3052	3380	3380
Harrisburgh, Pennsylvania	187	125	90	5097	5098	5375	5375
Hartford, Connecticut	273	262	202	5809	6168	6036	6036
Hatteras, North Caro'ina	(a)	9	0	(a)	2407	2571	2571
Havre, Montana	324	409	369	8601	7470	8556	8700
Helena, Montana	342	458	402	7887	7058	7663	7898
Houston, Texas	0	0	0	1489	1188	1157	1157
Huron, South Dakota	353	366	267	7945	7683	8004	8004
Indianapolis, Indiana	188	161	59	5172	4987	5298	5298
Jackson, Mississippi	5	2	(a)	2233	2075	(a)	(a)
Kansas City, Missouri	127	136	6	4869	4362	4956	4956
Kewanee, Illinois	224	267	133	6041	5800	6129	6139
Knoxville, Tennessee	89	36	0	3655	3354	3670	3670
La Crosse, Wisconsin	402	315	183	7648	7417	7322	7322
Lander, Wyoming	350	498	428	7785	7404	7812	7947

[1]Figures in this column are normal totals for a complete heating season, September to June, incl.

Figures in this table, with nine exceptions, based on local weather bureau reports. Exceptions are Utica, Lewiston, Kewanee, Sioux City, and Jackson, figures for which are furnished through the courtesy of Coke Sales Department, Central New York Power Corp., Utica, N. Y.; Norman E. Ross, Bursar, Bates College, Lewiston, Me.; J. M. Hartman, Engineering Department, Kewanee Boiler Corp., Kewanee, Ill.; Alfred R. Wagstaff, Engineer, Sioux City Foundry and Boiler Co., Sioux City, Iowa, and H. M. Ludlow, Box 1368, Jackson, Miss., respectively; Anaconda, Bozeman, Butte and Livingston, Mont., through the courtesy of the Montana Power Company.

[Table concluded on Page 120]



No. 44 Mills Water Tube Boiler

"There's a long, long trail . . . ."

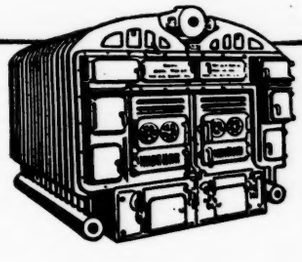
The hot flue gases in an H. B. Smith Cast-Iron Boiler rise directly to the highest surface of the boiler, across to each side, to the front through the upper flue passages, then to the back through lower channels, and finally up the chimney. It's really a long, long trail . . . and not a straight one.

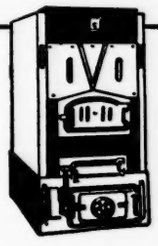
Actually, the gases would rather rush directly to the chimney, but the efficient "Mills" design hinders them at every turn. Instead, they're scientifically-directed along paths which give them a chance to scrub against maximum heating surface and give up more heat.

That's the secret of Smith-Mills efficiency. The result is more heat per fuel unit . . . less fuel used . . . lower fuel costs.

It *pays* to specify H. B. Smith boilers for industrial, commercial, institutional and residential jobs. If you want to know more important H. B. Smith advantages, send for your free catalog.

Support the I=B=R consumer advertising program — it's your campaign.





H.B. Smith

CAST-IRON BOILERS

THE H. B. SMITH CO., INC., 63 Main Street, Westfield, Mass. Offices and Representatives in Principal Cities

Degree-Days for May, 1947 (Concluded)

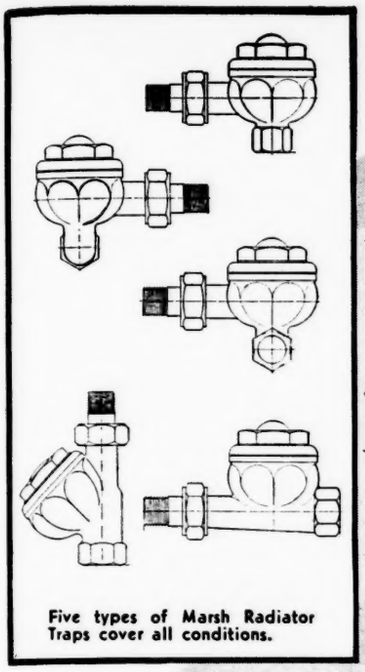
HEATING AND VENTILATING's 20th Year of Publication of Monthly Degree-Day Data

City	May			Cumulative, Sept. 1, 1946 to May 31, 1947			Heating Season[1]
	1947	1946	Normal	1946-47	1945-46	Normal	Normal
Lansing, Michigan	399	337	229	6781	6677	7048	7048
Lewiston, Maine	406	359	344	7348	7703	7662	7707
Lincoln, Nebraska	198	218	99	5855	5185	5999	5999
Little Rock, Arkansas	25	32	0	3102	2853	2811	2811
Livingston, Montana	365	522	405	7489	6748	7043	7205
Los Angeles, California	67	126	90	1335	1417	1504	1504
Louisville, Kentucky	137	89	0	4277	3926	4180	4180
Lynchburg, Virginia	108	107	0	4158	4057	3980	3980
Macon, Georgia	22	7	0	2286	2176	2201	2201
Madison, Wisconsin	391	291	236	7149	6993	7429	7429
Marquette, Michigan	650	528	496	8369	8011	8464	8693*
Memphis, Tennessee	37	29	0	3152	2973	2950	2950
Meridian, Mississippi	14	3	0	2347	2111	2160	2160
Milwaukee, Wisconsin	474	385	350	7170	6808	7206	7245
Minneapolis, Minnesota	359	320	236	7888	7666	7850	7850
Modena, Utah	216	338	350	6162	6226	6520	6562
Montgomery, Alabama	5	0	0	1963	1925	1884	1884
Nantucket, Massachusetts	405	391	384	5523	5626	5834	5957
Nashville, Tennessee	102	56	0	3655	3398	3507	3507
New Haven, Connecticut	311	283	223	5640	5941	5895	5895
New Orleans, Louisiana	0	0	0	1219	1108	1024	1024
New York, New York	216	154	171	4833	4885	5244	5274***
†Nome, Alaska	1269	1454	1443	12,272	12,342	13,058	14580**
Norfolk, Virginia	60	48	0	3069	3063	3350	3350
North Head, Washington	348	369	422	4508	4695	4693	5452**
North Platte, Nebraska	258	290	199	6350	5578	6366	6366
Oakland, California	144	266	205	2877	3125	2870	3143**
Oklahoma City, Oklahoma	58	58	0	3601	3133	3613	3613
Omaha, Nebraska	214	198	84	6142	5572	6131	6131
Oswego, New York	388	355	357	6528	6695	7025	7088
Parkersburg, West Virginia	177	116	53	4653	4463	4775	4775
Peoria, Illinois	263	231	121	6021	5793	6109	6109
Philadelphia, Pennsylvania	165	101	115	4405	4441	4720	4737***
Phoenix, Arizona	0	0	0	1278	1479	1405	1405
Pittsburgh, Pennsylvania	183	148	78	5353	4845	5235	5235
Pocatello, Idaho	207	379	332	6570	6789	6595	6655
Portland, Maine	426	415	366	7129	7543	7137	7218
Portland, Oregon	109	137	245	3827	3951	4379	4469
Providence, Rhode Island	243	206	251	5232	5461	6015	6015
Pueblo, Colorado	203	278	186	5872	5292	5514	5514
Raleigh, North Carolina	48	41	0	3104	3125	3234	3234
Rapid City, South Dakota	375	466	344	7611	6521	7088	7118
Reading, Pennsylvania	184	123	96	4861	4910	5389	5389
Red Bluff, California	14	29	(a)	2557	2716	(a)	(a)
Reno, Nevada	160	289	366	5525	5609	5802	5892
Richmond, Virginia	83	64	0	3615	3613	3695	3695
Rochester, New York	334	309	260	6450	6605	6732	6732
Roseburg, Oregon	154	186	267	4096	3956	4314	4428
Roswell, New Mexico	27	39	0	3479	3193	3484	3484
Sacramento, California	24	67	53	2677	2619	2653	2653
St. Joseph, Missouri	169	171	16	5257	4774	5161	5161
St. Louis, Missouri	115	112	0	4501	4136	4585	4585
Salt Lake City, Utah	133	252	236	5756	5640	5555	5555
San Antonio, Texas	0	6	0	1844	1514	1202	1202
San Diego, California	45	48	115	1340	1365	1639	1645
Sandusky, Ohio	274	232	183	5706	5571	6208	6208
San Francisco, California	231	293	264	2632	2655	2681	3264**
Sault Ste. Marie, Michigan	674	548	502	8943	8778	8950	9285**
Savannah, Georgia	6	0	0	1585	1581	1490	1490
Scranton, Pennsylvania	245	212	171	5806	5815	6129	6129
Seattle, Washington	163	166	316	4198	4150	4695	4934**
Sheridan, Wyoming	333	460	403	7643	6811	7888	8008
Shreveport, Louisiana	0	6	0	2272	2039	1938	1938
Sioux City, Iowa	293	267	164	7127	6545	6898	6898
Spokane, Washington	185	244	285	6229	6034	6274	6355
Springfield, Illinois	159	147	56	5213	4861	5373	5373
Springfield, Missouri	131	151	12	4682	4218	4428	4428
Syracuse, New York	300	299	267	6498	6619	6881	6893
Tacoma, Washington	210	203	332	4594	4543	4883	5181**
Terre Haute, Indiana	191	148	0	5130	4757	4872	4872
Toledo, Ohio	304	268	180	6204	6079	6077	6077
Topeka, Kansas	131	136	34	4983	4347	4969	4969
Trenton, New Jersey	186	127	81	4778	4875	4933	4933
Utica, New York	259	326	253	6003	7561	6796	6796
Valentine, Nebraska	340	391	273	7269	6357	7039	7039
Walla Walla, Washington	63	137	171	4582	4418	4808	4808
Washington, D. C.	121	77	25	4036	4028	4626	4626
Wichita, Kansas	114	136	0	4727	4113	4673	4673
Williston, North Dakota	440	447	362	9184	8744	9269	9323
Winnemucca, Nevada	129	304	324	5790	5894	6274	6427**
Yakima, Washington	71	81	186	5471	5039	5599	5599

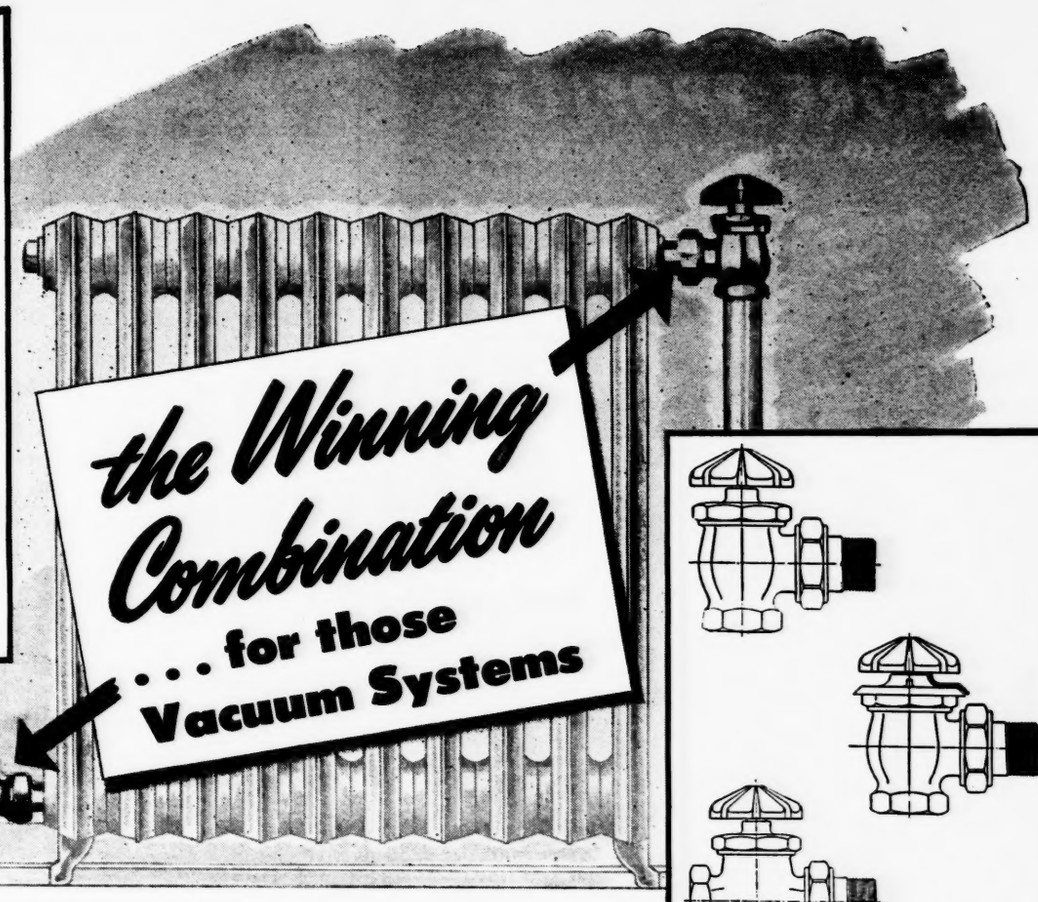
(a) Data not available.
†Nome data are for April.

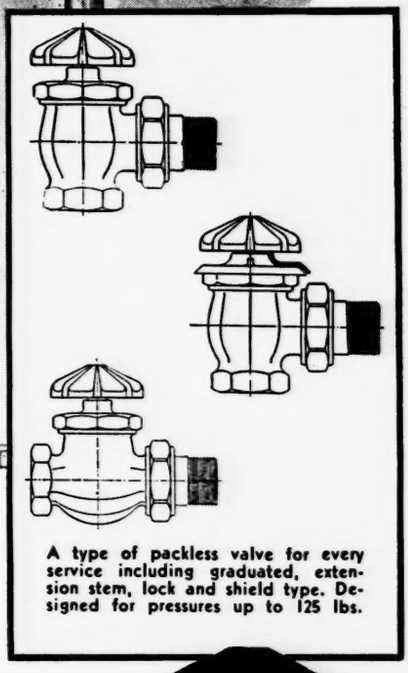
*Includes August.
**Includes July and August.
***New 48-year normal covering 1898 to 1946.

[1]Figures in this column are normal totals for a complete heating season, September to June, incl.



Five types of Marsh Radiator Traps cover all conditions.





A type of packless valve for every service including graduated, extension stem, lock and shield type. Designed for pressures up to 125 lbs.

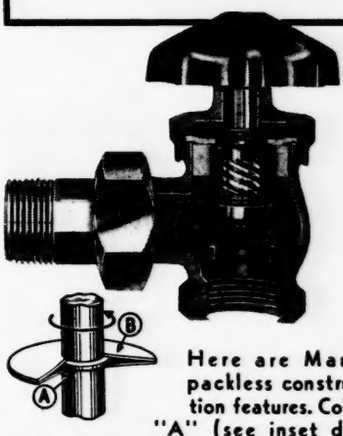
HERE is a team of thoroughbreds that will make every radiator function at its top efficiency in your steam or vapor systems—the combination that wins . . . against trouble, leakage, time.

The trap is a masterpiece of precision—the kind that's always positive. Its distinguishing feature is a diaphragm highly sensitive to narrow differences in temperature between steam, water and air at operating levels. As a result of this fine differentiation, steam is retained until it gives its full heat, yet all condensate, air, and gases are positively passed into the return system. Note the excellent construction, attractive appearance and wide range of patterns shown in the accompanying illustrations.

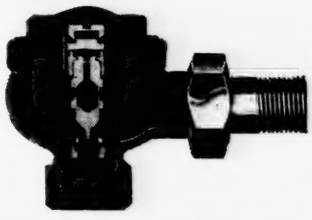
Worthy teammate of the Marsh Radiator Trap is the packless radiator valve. In too many valves of this kind the word, "packless", is lightly used to describe a valve that merely requires less packing than the conventional packed valve, but this is not one of those cases. The Marsh has no packing whatever. Instead it has a metal-to-metal seal. It never blows out along the stem; never leaks; is practically indestructible. Note the diagram of this packless feature opposite. Note, also, the finer construction throughout.

Both trap and valve are individually tested under actual steam operation. They are typical of a full line of quality traps, vents and other heating specialties that always do their job right. Ask for complete facts about Marsh products and engineering service.

MARSH HEATING EQUIPMENT COMPANY, 2059 Southport Avenue, Chicago 14, Ill.
Division of Jas. P. Marsh Corp.



Here are Marsh packless construction features. Collar "A" (see inset diagram) is machined on stem. Stainless spring-steel wafer "B" is pressed against collar to form permanently leakproof annular seal. Cross section shows quality construction.



A highly sensitive diaphragm does the job in the Marsh Radiator Trap. Note sturdy construction.

MARSH SINCE 1865
Quality Heating Specialties

Genuine WAGNER

Fast Moving Electric Motor Repair Parts *Available Now!*

M-1—For "one-trip" jobs . . . as useful to the service man as his pliers and screwdriver, and almost as easy to carry.

The M-1 kit box is of heavy fiber, contains a variety of 61 of the motor repair parts most commonly needed for domestic appliance type motors up to and including ½ horsepower.

M-2—For your shop . . . a convenient and economical way to get wide coverage of repair parts.

The M-2 kit box is of sheet steel, contains a variety of more than 680 fast moving parts for domestic and industrial type repulsion-start induction brush-lifting and capacitor-start motors up to and including 3 horsepower.

M-1 and M-2 kits are available now at our 25 branches and 325 authorized service stations. Or if more convenient, order your kits by using the coupon below.

Wagner Electric Corporation
6463 Plymouth Avenue,
St. Louis 14, Mo., U. S. A.

Date.....

Please ship .....(Quantity) Motor Parts Assortment M-1 ($13.05 net each).....$.....
Shipping Weight 3 pounds

F. O. B. Point of Shipment.....(Quantity) Motor Parts Assortment M-2 ($169.33 net each).....$.....
Shipping Weight 46 pounds

Order No..... Total $.....

Company.....

Address.....

City.....Zone....State.....

M47-21A

Wagner ME Electric

NEW CATALOGS

Underground Conduit System

A four-page folder covers basic types of tile and cast iron conduit systems for underground steam, and hot water and oil lines. Engineering data and sizing charts are included. — *Ric-wiL Company, Cleveland, Ohio*.....306

Steam and Hot Water Specialties

A series of bulletins covers air vent valves, thermostatic traps, pressure reducing valves, inverted bucket traps, and outdoor thermostats for hot water heating systems.—*Hoffman Specialty Co., 1001 York St. Indianapolis 7, Ind.* 307

Boilers

Kewanee square-heat type R boilers are the subject of a four-page bulletin, catalog No. 88-4. Boilers are for oil, gas or stoker. Sizes and capacities are tabulated.—*Kewanee Boiler Corp., Kewanee, Ill.* 308

Cartridge Filters

Cartridge replacement type filters for oil burner, industrial fuel, gasoline, and miscellaneous services are shown with specifications and flow data in bulletin MK 3217, eight pages, two colors. Cartridges feature graded density and resinous impregnation.—*The Cuno Engineering Corp., Meriden, Conn.* 309

Nickel Steel Tubing

A 10-page illustrated bulletin No. TR-508, describes the mechanical and working characteristics, heat treating behavior and weldability of a newly developed 9% nickel tubing and suggests fields for application.—*The Babcock & Wilcox Tube Co., Advertising Division, 85 Liberty St., New York 6, N. Y.* 310

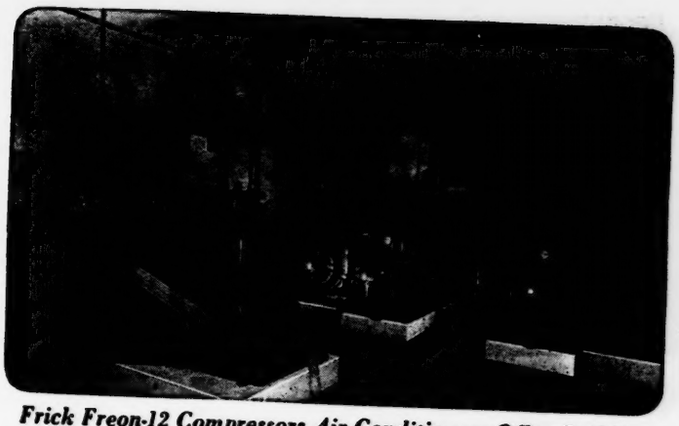
Refrigeration Units

Model WUA Freon-12 refrigeration condensing units are described in a twenty-four-page folder. Tables of sizes and notes on specifications are included. — *Worthington Pump and Machinery Corp., Harrison, N. J.* 311

Packaged Water Heating

An eight-page, three-color folder describes institutional and industrial water heaters of the direct-diffusion type which mix medium pressure steam (50 to 150 lb) with cold or warm water to provide an instant supply of hot water at controlled temperatures. Steam is used only when hot water is being drawn. Makers claim 40% more hot water per pound of steam. —*O'Brien Steam Specialty Co., 205 Harrison Street, Syracuse 2, N. Y.* 312

Frick Ammonia Compressors Serving a Large Hospital



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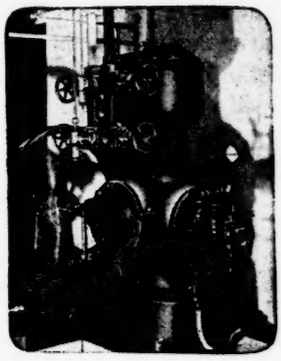

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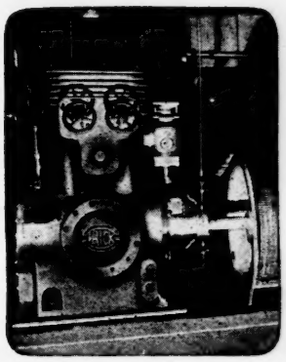

Frick Freon-12 Compressor

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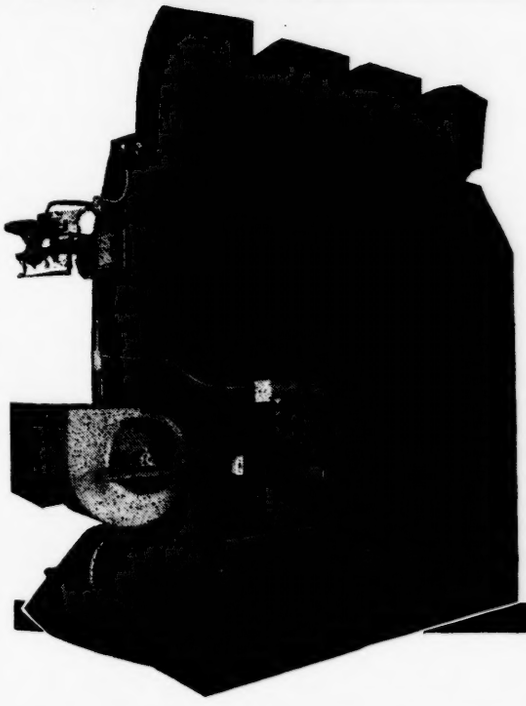

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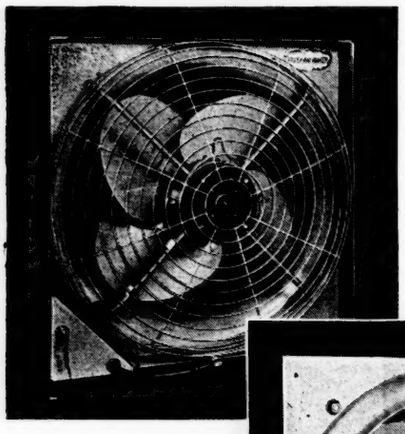

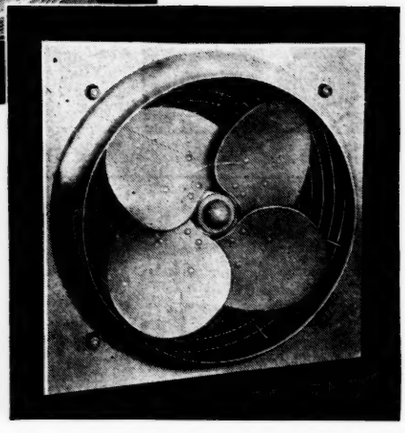

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A new bulletin, describing Flowrator instruments for measuring flow rate of liquids and gases includes descriptions and illustrations of instruments, both assembled and disassembled. Numerous new accessories are shown, including a horizontal line adapter which makes it possible to insert Flowrator meters as easily in horizontal runs as they previously could be inserted in vertical pipes. Details of front and rear panel mounting, rear lighting arrangements, steam heating coils, and alarm devices are among the items included.—*Fischer & Porter Co., Dept. 2J-C, Hatboro, Pa.* 313

Heat Loss Calculation

National Fitters Guide, described as a simple and accurate method of figuring radiation and selecting the boiler for steam and hot water residential heating systems, is offered for use in combination with a heat loss calculation sheet developed by the National engineering service department.—*National Radiator Co., Johnstown, Pa.* 314

Warm Air Heating

"Proved Practices in Installing and Servicing Warm Air Winter Air Conditioning Systems" is the title of a 12-page booklet. All phases of warm air heating are covered including generator operation, forced air circulation, register temperatures, etc. — *Bacharach Industrial Instrument Co., 7000 Bennett St., Pittsburgh 8, Pa.* 315

Controls for Hotels and Schools

Specialized electric and pneumatic tools for hotel and school installations are described in two eight-page folders (one for each class of application). Included are heat controls, generating controls, air conditioning and cooling controls related to special applications in schools and hotels.—*Minneapolis-Honeywell Regulator Company, Minneapolis 8, Minn.* 316

Automatic Heating

Three folders describe respectively gas burners, oil burners and stokers. Each folder is illustrated and gives construction and capacity specifications.—*Consolidated Industries, Inc., Lafayette, Ind.* 317

Air Distribution Panels

Multi-Vent panels for uniform, draftless air distribution in heating, ventilating and air conditioning are described in a 58-page, stiff-bound book. Panels are an assembly of control plate and frame, displacement valve, and perforated distribution plate receiving air supply from ductwork or ordinary plenum above the panel. Applications of multi-vent in railroads, commercial and industrial buildings, are described and illustrated. Installation techniques and design data are included—*The Pyle-National Co., 1334 N. Kostner Ave., Chicago 51, Ill.* 318

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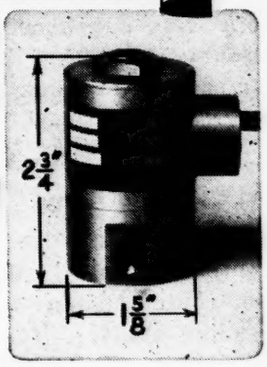

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Fans

A line of air circulating fans, including Sirocco, paddle wheel, exhaust and general duty propeller type fans are described in a four page bulletin with sizes, capacities and list prices.—*Circulators & Devices Mfg. Corp., 22 Rose St., New York 7, N. Y.*.....319

Air Conditioning Units

A 12-page three-color catalog covers two lines of air conditioning units, Latenaire and Sensaire, includes remote and self-contained models of both the floor and wall types as well as duct-pipe coils for use with any forced air system.—*American Coils, Inc., 360 Thomas St., Newark 5, N. J.* 320

Jacketed Cocks and Fittings

Bulletin 245 describes jacketed piping, fittings and valves for conveyance of viscous materials in the heated state. Photographs and drawings illustrate methods of assembling and heating.—*Parks-Cramer Co. 1105 Old South Building, Boston, Mass.*.....321

Refrigeration Controls

Controls for cooling and refrigeration, including temperature, pressure, defrosting time and motor controls are described and illustrated with rating tables and design data in a new catalog and price list.—*Minneapolis-Honeywell Regulator Co., Minneapolis 8, Minn.* 322

Try Cocks

Water column and gauge try cocks of various types for process and boiler applications, including steam pressures up to 2500 lb are the subject of Bulletin F, four pages, two colors.—*Ernst Water Column & Gage Co., Livingston, N. J.*.....323

Silver Brazing Alloy

A new bulletin, No. 15, describes Easy-Flo 45, a recently announced silver brazing alloy with lower silver content, lower operating temperature and fast spreading ability.—*Handy & Harman, 82 Fulton St., New York 7, N. Y.* 324

Sewage Disposal Equipment

Bulletin No. 23-120-B10 covers various types of equipment for use in sewage disposal plants and water works. Illustrations show rotary centrifugal installations of blowers and filters in a typical storage plant. —*Roots - Connersville Blower Corporation, Connersville, Ind.* 325

Timing Motors

Self-starting synchronous timing motors, type SX, 240 rpm, 60 cycle, are described in a four-page bulletin. —*The R. W. Cramer Company, Inc., Centerbrook, Connecticut.* 326

Catalog 444 Shows How to

CUT HEAT-UP TIME

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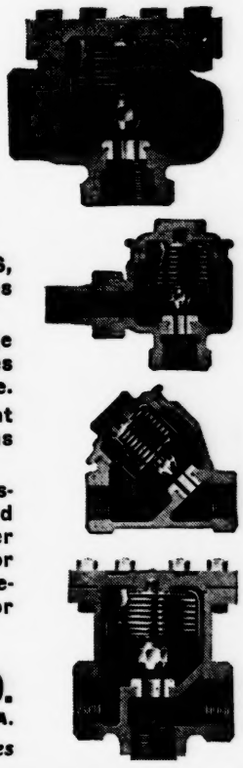
Unit Heaters and Pipe Coils



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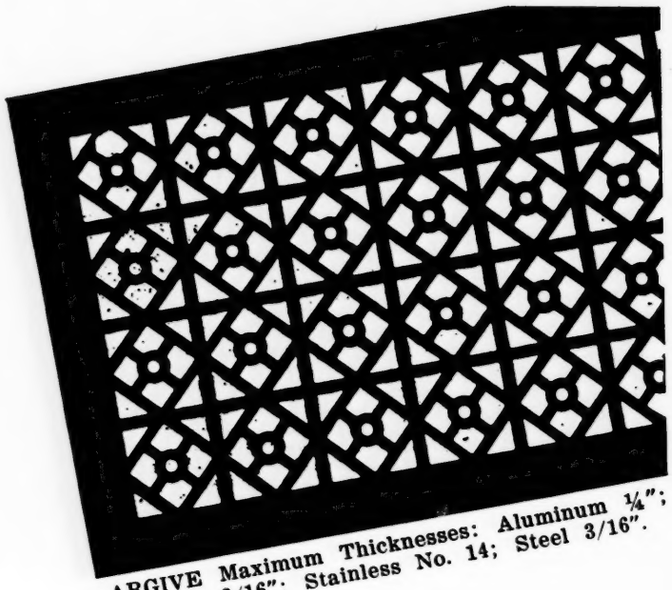

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Silicone Data Sheet

Preliminary data sheet lists specifications for rewinding air conditioning induction motors with silicone insulation. Parts of electric motors with the corresponding silicone requirements are listed. — *Dow Corning Corporation, Midland, Mich.* 327

Bitumastic Paint

Bitumastic Hi-Heat grey is described in folder No. 844A as a true aluminum paint changing to metallic grey at 700F. This is a specialized industrial protective coating, said to resist corrosion and high heat up to 1200F.—*National Marine Paint Co., Inc., 519 W. 47th St., New York 15, N. Y.*.....328

Heating Coils

Bulletin No. 147 gives physical data, friction tables, capacities and construction detail of heating coils.—*Chicago Fan and Fin Coil Co., 1937 W. Walnut St., Chicago 12, Ill.* 329

Strainers

Self-cleaning strainers for steam, air or liquid surface are described in Bulletin 465, eight pages, two colors. Illustrations show several typical strainer installations. Strainer screen interchangeability is stressed, and standard screen materials are listed for various applications.—*Leslie Co., 52 Delafield Ave., Lyndhurst, N. J.* 330

Condensing Unit Data

RCU 5000, Jack & Heintz Condensing Unit Data, is a condensation of essential service data for quick reference in sizing and installing condenser units. The manual includes an illustrated spare parts list, electric motor service stations and brief service data. —*Jack & Heintz Precision Industries, Inc., Cleveland 1, Ohio*.....331

JULY, 1947

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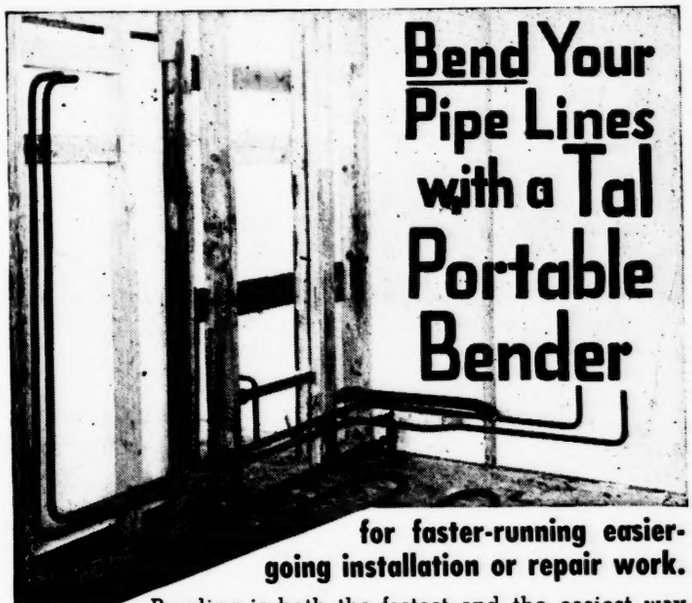

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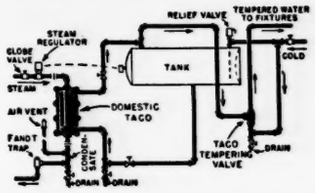

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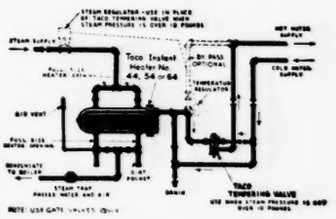

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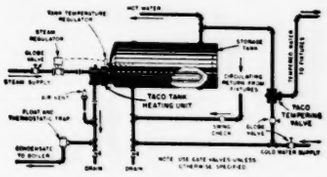

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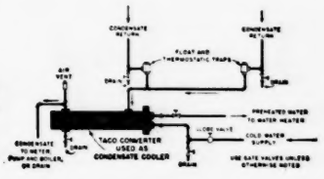

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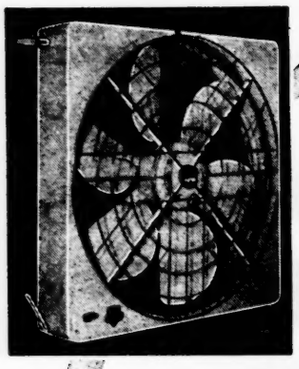

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Getting Personal

W. B. Harris

W. B. Harris *(Ventilation of Long Tunnels During Construction,* page 63) received his Ch. E. degree from Rensselaer Polytechnic Institute in 1933. From 1934-6 he was development engineer at the Dead Sea Works of the Palestine Potash Co., Ltd. In 1937 he received his M.S. degree from the Harvard Graduate School of Engineering and the same year became affiliated with the New York State Department of Labor, Division of Industrial Hygiene and Safety Standards.

Mr. Harris was with the Army Air Forces, Repair Depot at Hickam Field, T. H., from 1942-5, as Industrial Hygiene and Safety Engineer with the rank of Captain. He returned to his former New York position in 1945.

He is Chairman of Committee on Industrial Ventilation, American Conference Government Industrial Hygienists; member, Committee on Codes, of that association; member, Committee on Industrial Ventilation, ASHVE; member, Committee on Industrial Ventilation, American Public Health Association; member, Advisory Committee on Industrial Ventilation, International Labor Organization.

G. E. McElroy

G. E. McElroy *(Design of Injectors for Low-Pressure Air Flow,* page 81) was born in Pawtucket, R. I., on December 17, 1889. He received his M.S. degree in mining engineering from Lafayette College in 1912. Up to 1920, most of his experience was in mine safety work, including three years with the Bureau of Mines, two years with the New Jersey Zinc Company at Franklin, N. J., and one year as 1st Lt. C. W. S., at the American University Experiment Station, Washington, D. C. In 1920, he rejoined the Bureau of Mines and has served continuously in the Bureau's field service since that time, specializing in field research on mine and tunnel ventilation and mine air-conditioning. He is now supervising engineer of the Bureau's mine ventilation section.

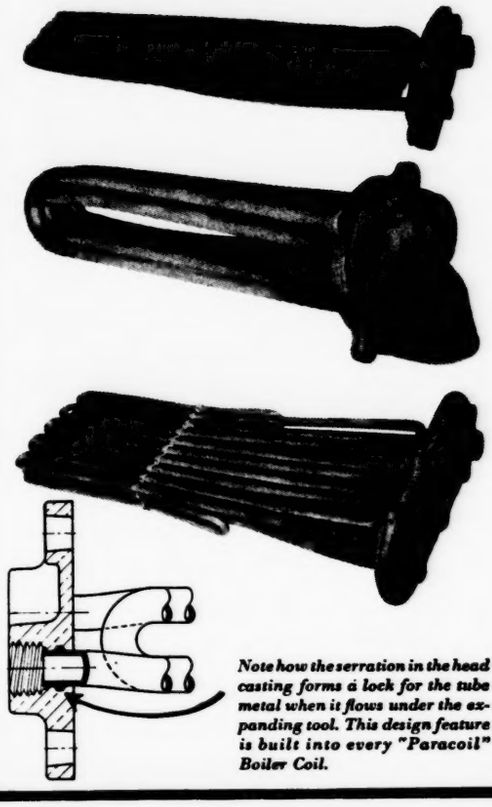
C. H. Neiman

C. H. Neiman *(Packaged Steam Generators,* page 66), Chief engineer of York-Shipley, Inc., York, Pa., is a charter member of the newly-formed Packaged Steam Generator Branch of American Boiler and Affiliated Industries. He was graduated from Lehigh University in 1935 with a B.S. in mechanical engineering and membership in two engineering honorary fraternities: Tau Beta Pi and Pi Tau Sigma. Entering York Corp. York, Pa., he moved from drafting to field erection to test plant and finally into development work with that organization. One of his important assignments there was the development of a portable room air conditioner. He joined the York-Shipley organization in 1942 as a development engineer, then assistant chief engineer, and finally chief engineer. With York-Shipley he has assisted in the develop-

Paracoil Submerged Boiler Coils

★ ★ ★ ★ ★ ★ ★ ★ ★ ★

for HOME and INDUSTRY



Note how the serration in the head casting forms a lock for the tube metal when it flows under the expanding tool. This design feature is built into every "Paracoil" Boiler Coil.

"Paracoil" has been building dependable boiler coils since 1915 for both home and industry. Here are a few of the varied types of instantaneous tankless water coils we are now fabricating for domestic residence boilers.

All head castings have SERRATED TUBE CONNECTIONS—another "Paracoil" design feature! You are doubly protected against leaks with a "Paracoil" Boiler Coil!

"Paracoil" Boiler Coils are designed for use with or without a storage tank. The larger industrial models are built with integrally cast water chambers and tube sheets to any desired capacity.

"Paracoil" manufactures many types of heating and ventilating accessories—all distributed through franchised agents. Some lucrative territories are still available. If interested, write our Elizabeth office for details.

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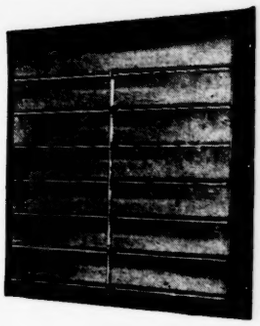
BUY OR SPECIFY "PARACOIL"

Elgo Ventilating Specialties

A Superior Automatic Shutter

An automatic shutter that not only opens more quickly but also closes more tightly. Completely weather-stripped around the inner edge of the frame to make a snug, tight fit. Noted also for absence of blade flutter. And it has swivel joints that never get loose or rattle. Sizes from 8" to 72" square—also rectangular.

Write for circular and prices.



"ELGO" TYPE AUTOMATIC SHUTTER
Front View (Closed)

Free CATALOG

ELGO SHUTTER & MANUFACTURING CO.
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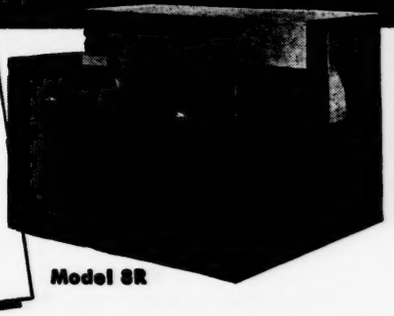
BRYANT SILICA-GEL AIR-DRYERS



Designed for efficient CO-ORDINATED OPERATION with all types of AIR CONDITIONING EQUIPMENT

Model 8R

• Provides a continuous flow of dry air offering accurate control of humidity in air conditioning systems for

PRODUCT FABRICATION
PRODUCT DRYING **HUMAN COMFORT**

AIR CONDITIONING ENGINEERS AND CONTRACTORS
FOR COMPLETE DATA, WRITE TO
BRYANT HEATER COMPANY
AIR DRYER DIVISION, 17825 ST. CLAIR AVE., CLEVELAND 10, OHIO
ONE OF THE DRESSER INDUSTRIES

HOSE FOR USE IN ANY INDUSTRY!

LIGHT, STRONG, FLEXIBLE

FLEXAUST

SPIRAL-REINFORCED HOSE

Flexaust is suitable for a wide variety of suction and pressure uses in ventilation, dust collection and fume control. **SIZES 1¼ inches to 24 inches.** Write for full information on this all-purpose hose. Various Flexaust types, accessories, and other hose types also available:

BLOFLEX *Non-reinforced collapsible*
PORTOVENT *Ring-reinforced retractable*

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Dept. B1, 15 Park Row, New York 7, N. Y.
Branch Offices: New Orleans, La., San Francisco, Calif., Washington, D. C.
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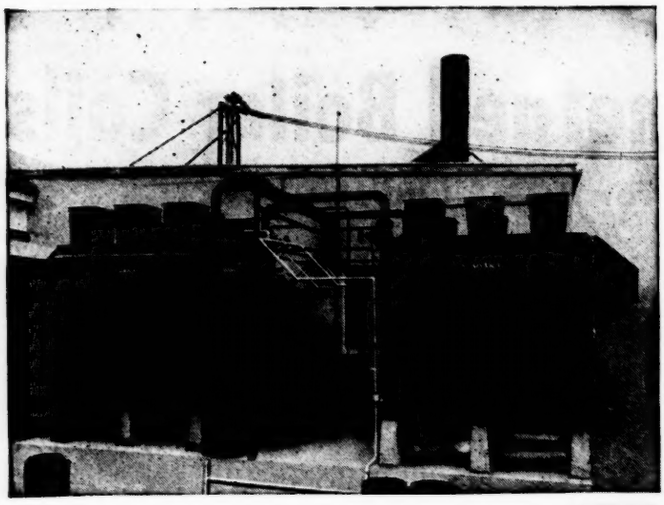

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TO INCREASE PRODUCTION

● YOU Can Obtain Greater Refrigeration Capacity from your Present Compressor by using NIAGARA DUO-PASS AERO CONDENSER.

This is possible because you use outdoor air as the cooling medium with the extra benefit of evaporative condensing! Condensing temperatures are lowered; condenser capacity is increased; head pressures are reduced —additional compressor capacity is the result.

Only the patented NIAGARA Duo-Pass makes this gain permanent because no other evaporative condenser lowers condensing temperatures to the point where deposits do not accumulate on condenser tubes. This does away with loss of time for cleaning and assures always full capacity.

Write for Bulletin 91-HV. and 103-HV.

NIAGARA BLOWER COMPANY

Over 30 Years of Service in Industrial Air Engineering

405 Lexington Ave., New York 17, N. Y.

Field Engineering Offices in Principal Cities

INDUSTRIAL COOLING HEATING • DRYING

NIAGARA

HUMIDIFYING • AIR ENGINEERING EQUIPMENT

ment of many new and unusual products such as an oil-fired shower-bath and sterilizer unit and mobile oil-fired laundry unit for the U. S. Army; a "packaged" oil-fired heating unit for veteran's housing projects; and various other residential and industrial oil heating units, not the least of which is the York-Shipley steam generator. He is a member of ASME, ASHVE, and the engineering committee of the Oil Heat Institute of America.

Michael S. Martus (Co-author, *Heat Transfer from Ground to Water in Buried Coils*, page 71), was born in New York City on February 28, 1925. Graduating from Woodmere (Long Island) High School in January of 1942, he worked as a page boy in a New York bank until July 1942, when he entered Antioch College. After completing 12 months of study and job experience at Antioch, he entered the Navy V-12 program at the University of Illinois in July 1943. Receiving his Bachelor of Science in Mechanical Engineering in October 1945, he was then commissioned as an Ensign in the Naval Reserve and served as Junior Engineering officer aboard the U. S. S. Atlanta. In August 1946, Mr. Martus was released to inactive duty, and in October of that year he entered Cornell University. While working for his master's degree, he was a part-time instructor in refrigeration and air conditioning. He expects to complete work for his degree in August, and will then work for the American Gas and Electric Corp.

W. F. King

M. S. Martus

William F. King (Co-author, *Heat Transfer from Ground to Water in Buried Coils*, page 71) was born in New Orleans on August 31, 1926. He graduated from High School in August, 1943; entered Tulane University in November, 1943, and after completing two semesters, was assigned to the Navy V-12 program at Tulane. After receiving his degree of Bachelor of Science in Mechanical Engineering, he was commissioned as Ensign in the Naval Reserve and subsequently released to inactive duty in July 1946. After attending Cornell University for one year, Mr. King received his Master of Science degree in Engineering in June 1947. His favorite hobby is flying. Mr. King expects to do heat pump development work around New Orleans.

INDUSTRIAL DEGREE-DAYS
May, 1947

City	55F Base	45F Base
Baltimore, Md.	29	0
Buffalo, N. Y.	136	42
Chicago, Ill.	125	13
Cleveland, Ohio	92	22
Detroit, Mich.	107	25
Indianapolis, Ind.	50	3
New York, N. Y.	63	6
Philadelphia, Pa.	41	4
Pittsburgh, Pa.	54	12
St. Louis, Mo.	15	0

Self-Cleaning Feature of

DEPENDABLE
Water Regulating Valve

. . . MEANS "TROUBLE-FREE" ACCURACY AND LONGER SERVICE

You'll avoid trouble—

due to dirt and lime in your water when you install the new A-P Model 68-B Water Regulating Valve on your water-cooled condensing units. A radically different construction principle actually "cleans itself," employing a molded composition seal on the valve stem which moves across the polished mirror-smooth stainless steel seat with an effective "wiping" action. Laboratory tested in sandy water, this unique sliding seal and seat showed no sign of water after the equivalent of many years use — proof of "lifetime" of accurate water regulating service.

With an operating head pressure of 50 to 150 p.s.i. Model 68-B can be easily adjusted to your system through the wide port in the body. Its long stroke provides closer, more uniform control without chattering or water-hammer. Three sizes, ⅜", ½" and ¾" inlet and outlet, offer a wide range of utility on Freon, Methyl Chloride or Sulphur Dioxide systems. Buy it today at your leading Refrigeration Jobbers — or write for bulletin No. M100.

DEPENDABLE Refrigeration Valves include: Thermostatic Expansion Valves, Refrigerant and Water Solenoids, Suction Pressure Regulating Valves, "Trap-Dri" Drier-Filter-Strainer and "Trap-It" Filter-Strainer, and Thermostats for cooling or heating.

AUTOMATIC PRODUCTS COMPANY

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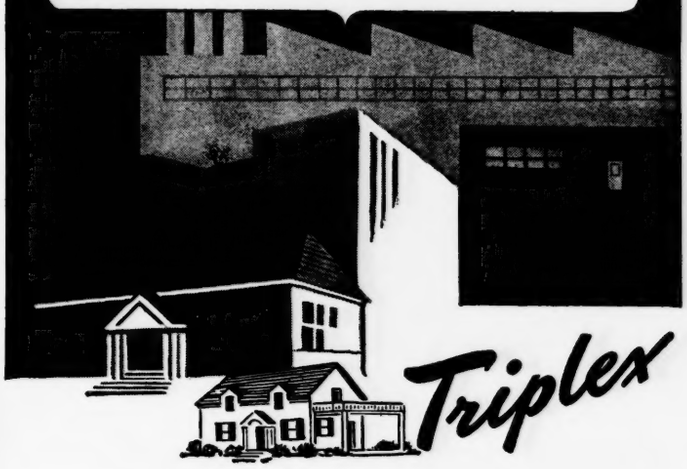
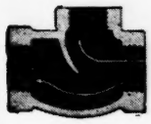
DEPENDABLE
REFRIGERANT VALVES
FOR AIR CONDITIONING • REFRIGERATION • HEATING

FOR HOMES AND BUILDINGS of any type—large or small



Triplex

FLOW CONTROL UNITS
Assure Hot Water at its Best



One-Pipe Distributors

Relief and Pressure Reducing Valves

Flow Control Valves

Direct Drive Circulators

Because the entire Triplex Flow Control system is so effectively simple, the same equipment that assures hot water heating at its best in homes can also be applied to large installations (stores, offices, public buildings, factories, etc.) by merely increasing the number or size of individual operating units. Hot water is automatically circulated—abundant heat supplied to the radiators instantly, whenever and wherever needed. As a result, the amount of radiation needed for unvarying comfort in any season is greatly reduced—and heat loss (through circulation) is reduced to an absolute minimum. Yes, Triplex means hot water at its best—in performance, popularity, and profits. Write for latest catalog today.

Triplex Sump Pumps—designed and built for long, trouble free service—are always in demand, too.

Triplex

HEATING SPECIALTY CO., INC.
PERU, INDIANA

In addition to units illustrated, the Triplex line includes—Direct Drive Horizontal Circulators, Indirect Drive Circulators, Expansion Tanks, Air Eliminators, Water Mixing Valves, Air Vent Valves, Indirect Heaters, Tankless Heaters, Control Units, Electrical Controls, and Barometric Draft Dampers.

SEE YOUR WHOLESALER TODAY

We Didn't Want to Make Strainers

BUT we needed them for use with Yarway Steam Equipment and couldn't find any to suit us.

So we designed the *YARWAY Strainer.*

Apparently a lot of other people were having "Strainer trouble" because the Yarway Strainers *purchased* in a few years now number *many thousands.*

Why don't you look into this better Strainer for your money? Buy one from your supply house (nearly 150 Mill Supply Houses now have them).

Cadmium Plated for protection against corrosion and for better appearance.

Examine its *high grade Monel woven wire screen* that stops dirt —lets condensate or other fluids flow freely.

Notice the removable blow-off bushing. Screen and bushing come out together—go back together, *automatically aligning.*

Six standard sizes from ½" to 2" for pressures to 600 lb.

See your Mill Supply House or write for Bulletin S-201

YARNALL-WARING COMPANY
104 Mermaid Ave. PHILADELPHIA 18, PA.

YARWAY STRAINERS

CANADIAN DEGREE-DAYS FOR MAY, 1947*

City	May 1947	May Normal	Cumulative, Sept. 1 to May 31 1946-47	Cumulative, Sept. 1 to May 31 Normal
Calgary, Alta.	484	480	9456	9111
Charlottetown, P. E. I.	477	536	7771	8263
Crescent Valley, B. C.	291	403	7278	7731
Edmonton, Alta.	453	428	10,535	9797
Fort William, Ont.	651	567	10,159	10,045
Grande Prairie, Alta.	465	505	11,015	10,122
Halifax, N. S.	450	493	6862	7395
London, Ont.	440	307	7206	7275
Medicine Hat, Alta.	384	301	9313	8495
Moncton, N. B.	459	474	8133	8568
Montreal, P. Q.	435	325	8163	8413
North Bay, Ont.	601	434	9568	9180
Ottawa, Ont.	452	310	8566	8674
Penticton, B. C.	194	273	6423	6346
Porquis Junction, Ont.	794	558	11,013	11,144
Prince George, B. C.	471	490	8996	8996
Quebec City, P. Q.	549	428	8785	9276
Regina, Sask.	539	434	11,471	10,891
St. John, N. B.	477	505	7422	8081
Saskatoon, Sask.	552	397	11,658	10,478
Toronto, Ont.	415	341	6750	7236
Vancouver, B. C.	257	326	5417	5303
Victoria, B. C.	273	366	4864	4935
Windsor, Ont.	341	251	6441	6705
Winnipeg, Man.	583	397	10,987	10,841

*These data are supplied through the courtesy of the Meteorological Division, Air Services Branch, Department of Transport, Canada, with the cooperation of N. W. Etter, Heating Engineer, Enamel & Heating Products Limited, Sackville, N. B.

. . Since the Last Issue

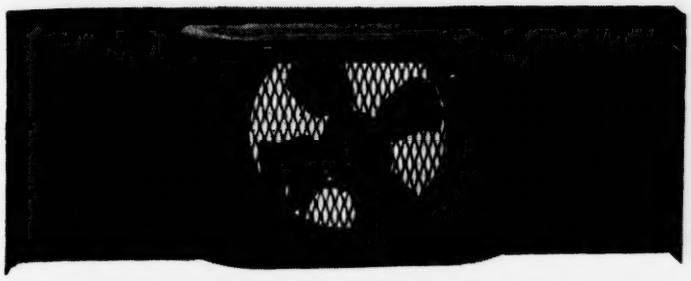
James H. Binger has been elected assistant vice-president of the **Minneapolis-Honeywell Regulator Co.** A graduate of the University of Minnesota law school, Mr. Binger had been assistant secretary of the company, since 1945, from which position he has resigned to assume his new duties. Formerly with the law firm of Dorsey, Colman, Barker, Scott & Barber, of Minneapolis, Mr. Binger joined Honeywell in 1943 to assist, in legal and contractural work in connection with the company's war work. In his new duties, he will devote most of his attention to the sales department and will assist Thomas McDonald, vice-president in charge of sales.

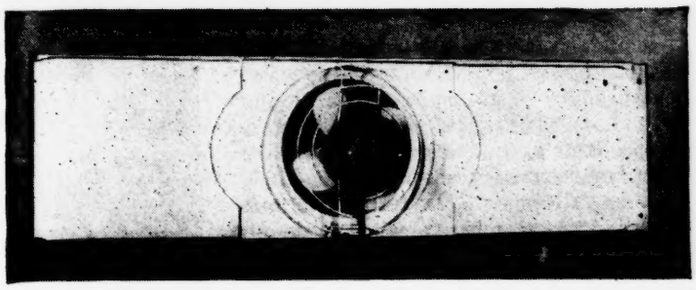
J. H. Binger

An entirely new building is being built as an addition to its factory by **Acme Industries, Inc.**, at Jackson, Michigan. The new building, 100 x 150 ft will be devoted exclusively to the increased production of Acme's new models of evaporative condensers, which were shown at the Cleveland Exposition, and floor type unit coolers.

Selection of **T. Keith Glennan**, 41, now an executive of Ansco Division of General Aniline & Film Corporation, Binghamton, N. Y. and wartime director of the U. S. Navy Underwater Sound Laboratory, as president of **Case Institute of Technology** has been announced. Mr. Glennan will be the first business executive to head the Cleveland engineering school, which has had but three presidents since its establishment 67 years ago as Case School of Applied Science and which adopted its new name on July 1st. He succeeds Dr. William E. Wickenden who retires on September 1st, after 18 years of service.

SPECIAL INTRODUCTORY OFFER!





YOU PAY	$17.50
YOU SELL 10" FAN FOR . . .	35.00
YOU SELL 6" FAN FOR . . .	11.50
TOTAL SELLING PRICE	$46.50
YOUR PROFIT	$29.00

10 INCH ADJUSTABLE

VENT FAN

PLUS 6 INCH

VENTILATING FAN—*Free!*

50% OFF

Take advantage of this special offer. Order 10" Adjustable Vent Fan for $17.50 (50% off $35 list price) and get 6" Ventilating Fan ABSOLUTELY FREE.

10" Fan is adjustable from 24" to 36", has smooth baked enamel finish on heavy gauge steel, General Electric blades, General Electric lifetime oiled motor, heavy rubber cord, rubber plug and pull chain switch. 6" Fan is adjustable from 24" to 36", has baked white finish, fully enclosed motor, wrapped and varnished windings, self-aligning bearings. ORDER THIS SENSATIONAL OFFER TODAY! ORDERS SHIPPED SAME DAY AS RECEIVED!

STANDARD ELECTRIC
MANUFACTURING CO., INC.
WEST BERLIN, NEW JERSEY

Yes - Gas Fired REZNOR HEATERS

with the **BIG, QUIET FAN**

Floor models or suspended type Reznor gas heaters economically provide heating comfort for offices, factories, and commercial establishments. Easy to install, low maintenance, and less operating trouble make these heaters first choice for occupants and property owners. We will gladly send you literature, write today.

REZNOR
Gas fired unit heaters

SUSPENDED OR FLOOR MODEL

A SIZE FOR EVERY NEED

REZNOR CO. MERCER, PENNA.

GAS FIRED HEATERS SINCE 1888

NO BOILERS • NO STEAM LINES
NO FUEL STORAGE • NO FIRE TENDING

Something New in Fan Bulletins

Bulletin 471 contains complete performance tables on Type ME Fan together with useful engineering data. It also illustrates and describes the new and improved PL (Power Limiting) Wheel which operates at medium speeds and higher efficiency for a wide range of application.

Your copy sent promptly upon request

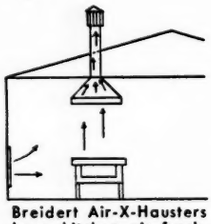
The NEW YORK BLOWER COMPANY
GENERAL OFFICES
3165 SOUTH SHIELDS AVENUE • CHICAGO 16
FACTORIES AT LA PORTE, IND. AND CHICAGO, ILL.

Breidert Air-X-Hausters

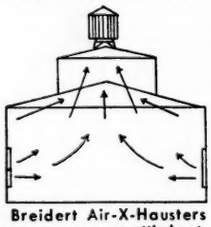
PAT. NO. 2269428

SAFE, SURE VENTILATION

NO MATTER WHICH WAY THE WIND BLOWS



Breidert Air-X-Hausters keep kitchen air fresh.



Breidert Air-X-Hausters remove summer attic heat.

Breidert chimney tops prevent smoke filled rooms.

Breidert Air-X-Hausters have a patented baffle construction which provides positive ventilation regardless of wind direction. Breidert flue caps keep heater pilot lights lit. Breidert Air-X-Hausters remove kitchen odors and attic heat. Breidert chimney tops keep rooms free of obnoxious fumes and smoke.

The far greater efficiency of Breidert Air-X-Hausters also means better ventilation for barns and chicken houses. Breidert Air-X-Hausters are neat and compact—attractively designed. Adaptable to all types of structures.

Certified tests by Smith, Emery & Co. prove the truth of the slogan "the greatest scientific improvement in ventilators in more than fifty years!" You specify *certified* performance when you specify Breidert Air-X-Hausters.

Send for the new Breidert engineering data book. It gives complete specifications on all Breidert Air-X-Hausters.

"Ask to see this demonstration."

G.C. BREIDERT CO.

3129 SAN FERNANDO ROAD

LOS ANGELES 41, CALIFORNIA

Louis D. Striebel of Youngstown, Ohio, resigned as manager of the Lombard Corp. of that city to become manager of the Michigan City plant of **C. A. Dunham Co.** He assumed his new duties June 2. Mr. Streibel, who spent his boyhood and high school years in Michigan City, attended Purdue University and Massachusetts Institute of Technology prior to entering his country's service in World War I. During World War II he was in charge of the shell manufacturing division of Pullman Standard Car Mfg. Co. Other connections in related fields were with the Skidmore Pump Co., the Economy Pump Co. and some nationally known industrial engineers.

Harold V. Engh, executive vice-president of Anaconda Wire & Cable Co., New York, has been elected president and general manager of **The Pyle-National Co.** Mr. Engh is a native of Chicago and a product of its public schools and of Lane Tech. In 1928 he became president of Inland Cable Co. When Inland consolidated with Anaconda Wire & Cable Co. in 1929, Mr. Engh joined Anaconda as vice-president. The new president of Pyle-National is a member of the board of directors of both Turner Brass Works and the National Bank & Trust Co. of Sycamore, Ill., as well as Anaconda Wire & Cable Co. **Other officers of The Pyle-National Co.**, elected with Mr. Engh are **A. N. Martin**, vice-president; **John L. Vogel**, secretary-treasurer; and **Ella M. Franklin**, assistant secretary-treasurer.

H. V. Engh

C. Thomas Miller has been named eastern regional manager of appliance sales for **Rheem Mfg. Co.** and will make his headquarters in New York. He replaces W. W. Stevens who has resigned. The appointment of **Reese B. Lloyd** as manager of Rheem's western plants has been announced. He will make his headquarters in Los Angeles.

The **Hendrick Mfg. Co.** of Carbondale, Pennsylvania, producers of perforated and fabricated metals, announces the appointment of **H. H. Buckley** as its Pittsburgh district manager, filling the post left by the recent death of Bruce G. Shotton. The offices of the Company will be maintained in the Gulf Building, Pittsburgh.

Trion, Inc., McKees Rocks, Pa., formerly Pittsburgh Range Co., announces the appointment of **John W. Frazier** as general sales manager. Mr. Frazier has been identified with Equitable Gas Co., Pittsburgh since 1936; serving successively as house heating salesman, architect and builder representative, residential heating supervisor, and supervisor of dealer relations. As general sales manager, Mr. Frazier will direct all sales and sales promotion activities of Trion who are building an electrostatic precipitator for homes and commercial establishments.

J. W. Frazier

Champion Coal Co., an affiliate of Pittsburgh Consolidation Coal Co., has accepted a franchise to sell and install Winkler domestic, commercial and industrial stokers in the Pittsburgh area, according to a joint announcement made by **U. S. Machine Corp.** which manufactures the Winkler line at Lebanon, Ind.

J. A. Ariano, 136 Liberty Street, New York, N. Y., has been appointed representative for **Illinois Testing Laboratories, Inc.**, 420 N. La Salle Street, Chicago. Mr. Ariano will represent the complete line of Alnor instruments on an exclusive basis. He succeeds Walton & Torrey, former representatives.

Minneapolis-Honeywell Regulator Co. has opened a new branch office at 1357 N. Street, Fresno, Calif., to service the San Joaquin Valley area. **Clyde N. Bradley**, recently in the San Francisco office of Honeywell, has been appointed district representative of the Fresno area.

Clifford M. Sayre has been appointed general works manager of the **Sturtevant Division of Westinghouse Electric Corp.** In this position Mr. Sayre will direct manufacturing operations at the Division's plants at Hyde Park, Mass., Camden, N. J., La Salle, Ill., and Berkeley, Calif. Mr. Sayre goes to Hyde Park from Sunnyvale, Calif., where he was works manager of the Joshua Hendy Iron Works. The new works manager is returning to Westinghouse after an absence of seven years. He joined the company originally in 1922 at the Springfield, Mass., plant, and served there and at the Lima, Ohio, plant for 18 years as production supervisor, superintendent, order service manager, and production manager. He left in 1940 to become vice-president and general manager of the Enterprise Engine and Foundry Co. at San Francisco, Calif. In 1943, he became associated with the Hendy Iron Works.

Frank P. Maxwell has been elected president and treasurer of the **Crise Mfg. Co.**, Columbus, Ohio, succeeding William C. Lerch, who is retiring because of ill health. Controlling interest in the electrical manufacturing concern has been acquired by W. F. Rockwell, Jr., president of the **Rockwell Mfg. Co.**, Pittsburgh; H. C. Stuckeman, general manager of Delta Mfg. Division of the Rockwell company in Milwaukee, and Mr. Maxwell, who has been assistant to the vice-president of sales of the Rockwell company at Pittsburgh. The Crise company's principal product has been an automatic furnace heat control which inc'udes patented shaded pole, reversible electric motor and thermostat and surfacestat of exclusive design.

F. P. Maxwell

C. W. Henderson has joined **Clarostat Mfg. Co., Inc.**, manufacturers of resistors, controls and resistance devices, as Sales Engineer in the Philadelphia area. For the past thirteen years he has been with RCA, holding various positions in the manufacturing and engineering activities of the RCA Victor Division.

H. C. Wallace has been appointed manager of **Air Reduction's** Louisville District. Mr. Wallace succeeds R. S. Moore, recently deceased.

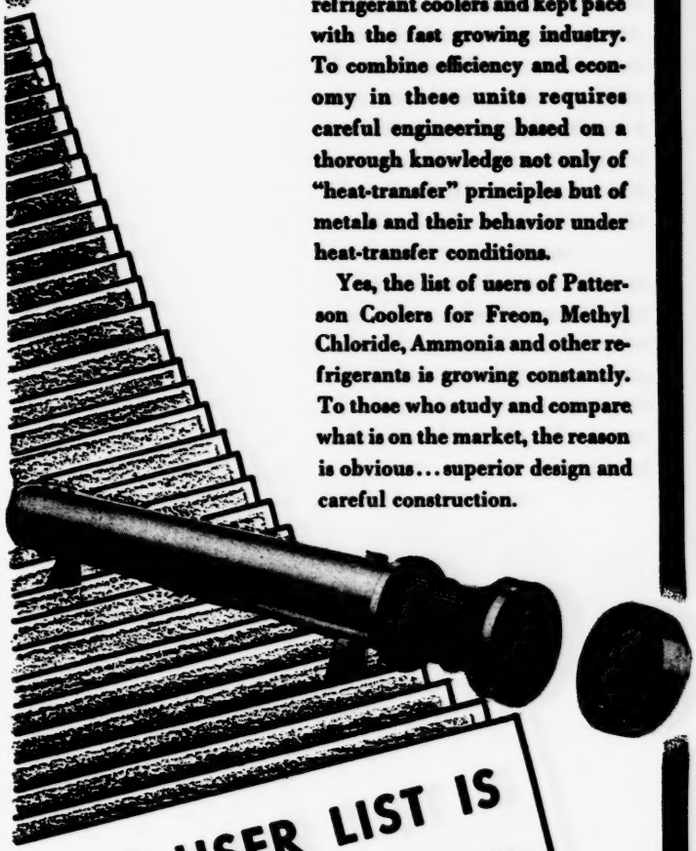
Sterling, Inc., manufacturers of heating specialties, controls, and pumps has been purchased by a group of Milwaukee business men. Two of the new owners will be active in the management of the company. **Paul A. West**, with a background of over twenty years' experience in sales work for some of Milwaukee's leading metal products manufacturers, will direct the sales along with his duties as president of the new company. **John B. Ballard**, new vice-president in charge of manufacturing, comes to Sterling from a large Milwaukee manufacturer of electrical and automotive products.

Coolers

... by PATTERSON-KELLEY

Being "heat-transfer" specialists, it is only natural that long ago our engineers began to work on refrigerant coolers and kept pace with the fast growing industry. To combine efficiency and economy in these units requires careful engineering based on a thorough knowledge not only of "heat-transfer" principles but of metals and their behavior under heat-transfer conditions.

Yes, the list of users of Patterson Coolers for Freon, Methyl Chloride, Ammonia and other refrigerants is growing constantly. To those who study and compare what is on the market, the reason is obvious... superior design and careful construction.



THE USER LIST IS GROWING LONGER ... RAPIDLY

Write for new bulletin which describes the latest type of Patterson Dry Expansion Type Cooler illustrated above. You will be interested, for example, in the refrigerant chamber which is machined from a solid billet of rolled carbon steel with welded-in partitions.

THE PATTERSON-KELLEY Company, Inc.

122 WARREN STREET, EAST STROUDSBURG, PA.

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Representatives in Principal Cities

7 REASONS WHY
BINKS "ROTOJET"
SPRAY NOZZLES
GIVE YOU BETTER SERVICE

① **Clog-proof:** Rotojet nozzles are designed to be self-cleaning. ② **Better atomization:** The high centrifugal force developed by the flow of liquid through the nozzle breaks up the water into a perfect spray. ③ **The correct size:** Rotojet nozzles are made with orifices from 1/16 inch to 1 13/16 inches. There is a correct size for your installation. ④ **No moving parts:** There is nothing to get out of order. ⑤ **Longer life:** Extra metal at points of severe wear gives Rotojet nozzles long life. ⑥ **Simplicity of Design:** Rotojet nozzles are highly efficient and easy to install. ⑦ **Satisfaction guaranteed:** Binks guarantees every Rotojet nozzle.

There is a Rotojet nozzle for every spray job

SEND NOW FOR BULLETIN 450 describing these Rotojet nozzles in full

Binks MANUFACTURING COMPANY
3114-40 CARROLL AVE., CHICAGO 12, ILL.
OFFICES IN ALL PRINCIPAL CITIES

Thomas Coyle, manager of the **Du Pont Co.'s** Chlorine Products Division, retired May 31 after more than 40 years in the chemical industry. The company also announced that the Chlorine Products Division and Solvents Division would be merged into the Chlorine Products Division headed by **C. B. Shepherd,** now manager of the Solvents Division.

Frank A. Stortz, Jr., former heating appliance sales representative in San Francisco, has been appointed sales manager of the **General Electric Co.'s Fan Division.** An engineer graduate of the University of New Mexico, Mr. Stortz joined the Company in Fort Wayne, Ind., in 1930. He was subsequently connected with the publicity department in Schenectady, N. Y., and was a district representative in the Chicago and New York offices. In 1938 he was appointed a sales engineer for the Federal & Marine Division in New York, and seven years later went to San Francisco as appliance representative.

Petroleum Heat and Power Co., Stamford, Conn., announced that **Messrs. G. L. Rowsey, Harris A. Melasky,** and **Glenn Forgan** have been elected to the company's board of directors. Mr. Rowsey and Mr. Melasky are president and vice-president, respectively, of Taylor Refining Co., Taylor, Texas, whose capital stock was recently acquired by Petroleum Heat and Power Co.

Leslie Co., Lyndhurst, N. J., manufacturers of pressure and temperature regulators and controllers, strainers and whistles announces the appointment of the following agents to handle industrial sales and service: **The Sherwood Bishop Company,** 413 Woodland St. (Manchester) Hartford, Conn.; **Harold W. Frederick Co.,** N. 213 Division St., Spokane, Wash.

Baker Ice Machine Co., Inc. announces the opening of its new factory at South Windham, Maine. The company will produce compressors and condensing units ranging in size from 1½ to 15 hp for both ammonia and Freon. Executive, sales and engineering staffs are now at South Windham and all general correspondence, except regarding renewal parts, should be addressed there. Renewal parts will continue to be handled in Omaha, Nebraska.

B. A. Johnson, who recently resigned from Minneapolis-Honeywell Regulator Co. has moved to Chicago to become active in administrative and consulting work. His immediate association will be with **Barney Olson, Inc.** and **Condensation Engineering Corp.** Barney Olson, Inc. is one of the oldest manufacturers' representatives and service organizations for gas equipment in the Chicago area. Condensation Engineering Corp. designs, manufactures and does application engineering on "Vitroliner" flues and chimneys. Both companies are located at 122 South Michigan Avenue where Mr. Johnson will make his headquarters.

D. J. Murray Mfg. Co., Wausau, Wis., announces that their Chicago office is now in Room 648, First National Bank Building, 38 South Dearborn Street, where **O. J. Malina,** sales manager of their heating and ventilating division, is in charge. The former address of their Chicago office was 43 East Ohio St.

The Chicago regional office of the **Minneapolis-Honeywell Regulator Co.** and its industrial division, the **Brown Instrument Co.,** has moved to 351 East Ohio St., occupies the entire top floor of the building, and provides 30,000 sq ft of floor space.

Eastern Industries, Inc. has purchased the buildings, machinery and product production rights of the McIntyre Company, Newton, Mass. As McIntyre Division, Eastern Industries, Inc., the newly acquired facilities will continue to manufacture the same line of pumps and fluid motors that it has in the past—including hydraulic, fuel, alcohol, water, air, and metering pumps. Key personnel of the McIntyre Co. will continue in their positions for the new division.

The **American Society of Lubrication Engineers** has formed a Northern California Section which will meet monthly in San Francisco. **Wilbur Deutsch,** 636 Potrero Ave., San Francisco 10, Calif., is chairman of the section, **H. G. [illegible],** vice-chairman and **L. W. McLennan,** secretary-treasurer.

A. B. Ritzenthaler, vice-president in charge of sales of the **Tappan Stove Co.,** Mansfield, O., was one of the three persons recently elected to serve on the board of directors. The other two members are **William R. Mabee,** plant superintendent and **Harold O. Dysart,** secretary.

COMING EVENTS

SEPT. 1-4, 1947—Fall meeting of The American Society of Mechanical Engineers, Hotel Utah, Salt Lake City, Utah. Ernest Hartford, executive assistant secretary, 29 W. 39th St., New York, N. Y.

SEPT. 8-12, 1947—Second National Instrument Conference and Exhibit, Stevens Hotel, Chicago. Further details from The Instrument Society of America, 1117 Wolfendale St., Pittsburgh 12, Pa.

DEC. 1-5, 1947—Annual meeting of The American Society of Mechanical Engineers, The Chalfonte-Haddon Hall Hotel, Atlantic City, N. J., Ernest Hartford, executive assistant secretary, 29 W. 39th St., New York, N. Y.

DEC. 8-10, 1947—Annual convention, American Society of Refrigerating Engineers, Hotel Traymore, Atlantic City, N. J. Acting secretary, M. C. Turpin, 40 W. 40th St., New York 18, N. Y.

JAN. 26-29, 1948 — Fifth All-Industry Exposition of the Refrigeration Equipment Manufacturers Association, at Cleveland Public Auditorium, Cleveland, Ohio.

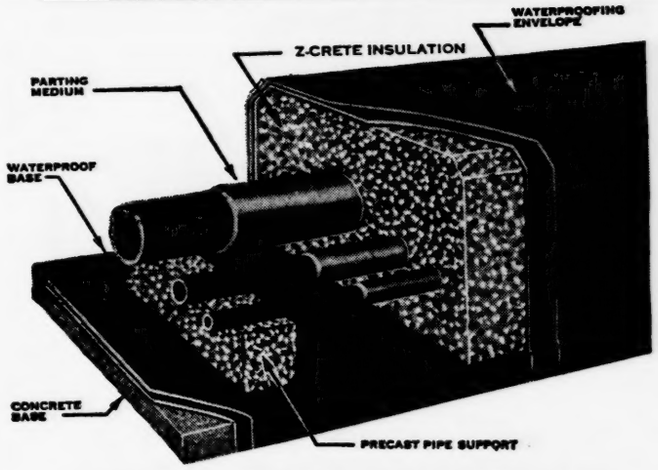
FEB. 1-4, 1948—54th Annual meeting, American Society of Heating and Ventilating Engineers, to be held in New York in conjunction with the 8th International Heating and Ventilating Exposition. Charles F. Roth, manager, International Exposition Co., Grand Central Palace, New York, N. Y.

JUNE 1-5, 1948—Annual meeting, Heating, Piping and Air Conditioning Contractors National Association, Hotel William Penn, Pittsburgh, Pa. Secretary, Joseph C. Fitts, 1250 Avenue of the Americas, New York, N. Y.

JUNE 9-11, 1948 — Spring meeting, American Society of Refrigerating Engineers, New Ocean House, Swampscott, Mass. Acting secretary, M. C. Turpin, 40 W. 40th St., New York 18, N. Y.

OCTOBER 6-8, 1947—Annual convention of the American Gas Association, Cleveland Auditorium, Cleveland, Ohio. Secretary, Kurwin R. Boyes, 420 Lexington Avenue, New York 17, N. Y.

"Z"-crete*
UNDERGROUND PIPE INSULATION
Permanent as the earth itself
Simple to install.



A New and Better Way to Insulate Underground Piping

Here are the big advantages Z-crete offers you: the Z-crete system differs from ordinary methods by approaching the insulation problem in a *new way.* It is cast in place *on the job,* forming a solid covering of water-repellent insulation, eliminating joints and voids which could fill with water. It is composed of permanent type materials which produce a rotproof, fireproof, moisture-resisting insulation. It will not disintegrate from wetting—nor from exposure to 1800 degrees F. It is as permanent as the earth itself!

Because Z-crete is applied four to six inches thick around pipes, it produces insulating efficiencies well in excess of ordinary practice. Pipes embedded in Z-crete Insulation are *supported continuously* and will *not* get out of line. Anchors are conveniently set in the structural base and the Z-crete conduit has ample compressive strength to carry heavy loads under traffic conditions without damage to the conduit!

Z-crete is *foolproof,* and *simple to install.* No foaming chemicals or special mixers are necessary. There are no prefabricated units which can be damaged from shipping or handling. Z-crete can be easily sawed or cut with a wood chisel and can be efficiently removed or patched.

Z-crete is by far the most flexible and dependable method of insulating underground pipes. It is always available, and you will find it *most* economical. Write us for full information on Z-crete—the new, better underground insulation—today!

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